Flow of Oil-Refrigerant 12 Solutions Through Horizontal Capillary Tubes.

By M. H. CHOUDHURY

Summary of Thesis.

The flow of Oil-Refrigerant 12 (Dichlorodifluoromethane) solutions through horizontal capillary tubes was studied. The effect of different concentrations of solution on the mass flow through tubes of different bores and lengths was the primary concern.

This is a new field of investigation; no previous work had been published on measurement of mass flows of such solutions. Although the capillary tube had been extensively used as an expansion device in refrigeration systems, specially in the smaller, and the hermetically sealed units, all investigations have been limited to the flow of R 12 only.

New experimental techniques had to be developed to measure the mass flow of oil - R 12 solutions, as none of the methods used by previous investigators on R 12 flow only, was suitable.

Interest in the flow of Oil - R 12 solutions arose from the lack of published data which might be applied to the design of the capillary tube as an oil return from the oil separator to the compressor crank case. This would provide a simple substitute for the usual, expensive and complicated float valve mechanism.

By/

By arranging the tubes as a by-pass in a main refrigeration circuit, complete control of the conditions was possible without having to establish a steady state in the main circuit. This improved accuracy, reduced the time required, so enabling a comprehensive series of tests to be conducted. The pressure and temperature distribution along the tube were also observed.

The analysis of the problem was hindered by the lack of adequate basic data on the properties of Oil - R 12 solutions at the relevant conditions. However, the mass flow of solutions has been compared with the actual and theoretical individual flows of R 12 vapour and oil.

The investigation provided the necessary data and showed that the capillary tube could be a simple and inexpensive substitute for the usual oil return device. Such a tube could now be designed to deal with compressor operating conditions where the maximum oil pumped over is known. At compressor conditions with smaller oil flows some loss of refrigerating capacity would be inevitable due to intermittent R 12 vapour leakage. It is shown that this loss is not large and that steps can be taken to reduce it.

UNIVERSITY OF GLASGOW

Flow of Oil-Refrigerant 12 Solutions through Horizontal Capillary Tubes and Application as an Oil-Return Device for Refrigerant Compressors

A Thesis Submitted for the
Degree of Master of Science

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By arranging the tube under test as a by-pass in a main refrigeration circuit, complete control of the conditions was possible without having to establish a steady state in the main circuit. This improved accuracy reduced the time required, enabling a comprehensive series of tests to be conducted. The pressure and temperature distributions along the tube were also observed.

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Chapter 2

INTRODUCTION

2. INTRODUCTION

The capillary tube as an expansion device is in general use in almost all small refrigeration units. It is particularly suitable for hermetic units. There have been several papers giving empirical data to aid the designer in selecting the appropriate size of capillary tube for a particular application. There have also been attempts at theoretically computing the mass flow of refrigerants through capillary tubes of varying length and internal diameter. The theoretical solution has not been achieved and empirical relations are still not wholly satisfactory to the designer.

In these investigations, care had been taken to eliminate lubricating oil from the refrigerant flow. Only one paper mentions the appreciable effect of the presence of oil (5). Almost all commonly used refrigerants dissolve varying quantities of lubricating oil when being pumped through the compressor. Some lubricating oil in the form of small drops is also pumped over by the compressor with the refrigerant vapour. In particular, R12 vapour is soluble in

refrigeration compressor lubricating oils, and liquid R12 and lubricating oil are completely miscible in all proportions.

In larger refrigeration units it is convenient to use oil separators placed between compressor and condenser to eliminate most of the oil from the refrigeration circuit, the oil being returned to the compressor crank-case through a ball or float-valve mechanism. The valve mechanism is necessary to account for the pressure difference between the compressor discharge and suction.

In smaller units the use of such oil separators has proved uneconomical. When an oil separator is not in use, it is of some advantage if the oil and refrigerant are miscible. The oil is then carried through the circuit by the refrigerant and returned to the compressor.

Besides its use as the expansion device in the main circuit, the capillary tube could also be used as an oil return line from the oil separator to the compressor crank-case, eliminating the need for an expensive and complicated float-valve mechanism. These two applications necessitate the study of the flow of oil-refrigerant solutions through capillary tubes

over a wide range of concentrations.

Information on the behaviour of such solutions in capillary tubes is virtually non-existent. The problem is so complex that an analytical solution is hardly feasible and could not be applied meaningfully because of the lack of basic data on the properties of oil-refrigerant solutions at the relevant conditions.

To keep the investigation within reasonable limits it was decided to experiment with one lubricating oil, sold under the trade name of 'Capella D', and dichlorodifluoromethane, commonly known by the trade name Freon 12, classified as Refrigerant 12 and abbreviated as R12, by the ASHRAE Standard (1).

New and more elaborate experimental techniques had to be developed to measure mass flows of oil-refrigerant solutions; the methods used by previous investigators, using refrigerant only, were not suitable.

While the primary interest has been a study of the mass flows of such solutions, comparison has also been made with the mass flows of oil and R12 vapour, individually. These later "end" concentration conditions are of interest also since under certain

conditions the capillary tube when used on the oil return to the compressor may be passing almost pure oil or refrigerant vapour only. Such a leakage of refrigerant vapour when the oil separator is not flooded may result in a loss of refrigerating capacity and the problem is discussed.

The large number of tests conducted provide a comprehensive series of results and a useful aid to the designer.

Chapter 3

NOMENCLATURE

3. NOMENCLATURE

Symbol	Meaning	Units
a	area of cross section	square inches
A	area of cross section	square feet
C, K	constant	dimensionless
d	diameter	inches
D	diameter	feet
f	friction factor	dimensionless
g	gravitational constant	32.2 ft/sec^2
H	enthalpy	British thermal unit (BTU)
J	mechanical equivalent of heat	770 ft-lb/BTU
m	rate of flow	lb/sec
M	rate of flow	lb/min
ml	unit of volume	milliliters
p	pressure	lb/in^2
P	pressure	lb/ft^2
psia	absolute pressure	lb/in^2 abs
psig	gauge pressure	lb/in^2 gauge
in	unit of length	inch, inches
ft	unit of length	foot, feet

Symbol	Meaning	Units
L	length	inches
L	length	feet
R_e	Reynolds Number	dimensionless
t	temperature	(^oF) degrees Fahrenheit
w	specific weight	lb/ft^3
W	total weight	lb
U	internal energy	BTU
v	velocity	ft/sec
V	volume	ft^3/ml
υ	specific volume	ft^3/lb
θ'	time	minute(s)
θ	time	second(s)
ρ	density	lb/ft^3
μ	absolute viscosity	lb/sec-ft
ν	kinematic viscosity	centistokes
press	pressure	
temp	temperature	
prefix Δ	difference	

Subscript		
	b	back pressure
	e	experimental
	g	H_2 vapour
	h	initial pressure

Subscript	i	individually
	l	mix liquid
	o	oil
	r	mix
	s	solution of oil-mix
	t	theoretical

(6) - Figure in brackets after a name refers to number used in bibliography.

Chapter 4

REVIEW

OF

PREVIOUS LITERATURE

4. REVIEW OF PREVIOUS LITERATURE

Capillary Tube Heat Exchangers

- R.E. Swart (33)

This is the first paper presented on the use of a capillary tube as an expansion device in heat-exchange with the suction line.

The concept is introduced of the "pump back" condition when a liquid seal is maintained at the capillary entrance. Heat exchange is advocated between the capillary tube and the suction line; this allows a larger quantity of colder and wetter refrigerant to reach the evaporator at the expense of spent suction vapour, increasing efficiency and capacity.

Two equations are given for determining the heat exchange length and the additional length using experimental constants.

Temp of warm liquid is assumed, t_{lf}

Temp of evaporator is assumed, t_e

Eff. of heat exchanger is assumed, e

(80 to 90 %)

Temp. $t_{f'}$ at which liquid starts to flash is determined

$$t_{f'} = t_w - 0.6\,e(t_w - v_o)$$

At temp $t_{f'}$ the saturation press $p_{f'}$ is obtained from the vapour pressure chart,

$$\text{then } L_1 = K_1 \frac{(p_w - p_{f'})^{0.5}}{q}$$

where L_1 = length in ft. of capillary tube required to reduce pressure from P_w to $P_{f'}$

K_1 = dimensionless constant

q = liquid flow rate lb/hr,

the additional length of tube required to reduce the pressure from $P_{f'}$ to the evaporator pressure is

$$L_2 = K_2 \frac{(P_{f'} - P_o)}{qV}$$

where L_2 = length required in ft.,

K_2 = dimensionless constant

V = sp. vol. of freon vapour at

P_o = evaporator pressure

total length of tube required is the sum of L_1 and L_2.

The constants K_1 and K_2 are determined

experimentally for each tube bore.

The method is not very satisfactory as prior to any other consideration, the experimental constants K_1 and K_2 have to be determined on the selected tube using the calorimeter and refrigeration already described.

The equations are not dimensionally balanced.

Theory and Use of a Capillary Tube for Liquid Refrigerant Control

L.A. Stoebler (31)

After enumerating the advantages of using a capillary tube, especially in hermetic units, Stoebler introduces the concept "capacity balance" between the condensing unit and capillary flow, indicated when a liquid seal is present at the tube entrance. Design based on what is called "Capacity Balance Characteristic" is advocated and the determination of and the factors influencing this characteristic are discussed.

It is stated that inclusion of any oil with the circulating refrigerant increases the viscosity of the refrigerant-oil solution and will have the effect of lowering the characteristic curve. However, neither argument nor experimental data is advanced in support of this statement.

Stoebler also advocates heat exchange with the suction line, for at least 40 inches length, at the condenser end of the capillary and states that overall capacity of the refrigerating system is increased by as much as 35%.

Graphs are presented of equivalent lengths of 0.042 in ID capillary tube against compressor displacement, for various evaporating temperatures, and two condensing temperatures.

The relationship is given to convert from one length and diameter of tube to another:

$$L_1 = L_0 \, (d_1/d_0)^{4.6}$$

where L_0 = length of capillary having inside diameter d_0

L_1 = length of capillary having inside diameter d_1

It is claimed that most of the conclusions are valid for R-22 also.

Applications and Characteristics of Capillary Tubes

H.F. Lathrop (16)

Lathrop discusses the flow of refrigerants through capillary tubes in simple language, and introduces the concept of "Capacity Balance", earlier advanced by Staebler (31), between the compressor and the capillary tube, and compares the system with hydraulic flow between two tanks, the condenser and the evaporator.

A simple graph is presented of compressor refrigerant pumping capacity for different head and suction pressures. A similar curve is presented of the flow capacity of a sample capillary tube under different head and suction pressures. The curves are superimposed to obtain capacity balance points; the designer then selects the conditions of operations.

The author presents a list of factors affecting the flow of refrigerants through a capillary tube, and those that have been considered.

Graphs are presented to show the variation of refrigerant circulation with the individual parameters of inside diameter, length of tube and pressure

difference. From these it is concluded that refrigerant flow

$$W \propto (D)^3$$

$$\propto \frac{1}{(L)^{\frac{1}{2}}}$$

$$\propto (\Delta P)^{\frac{1}{2}}$$

where W = weight of refrigerant circulation

D = inside diameter of tube

L = length of tube

P = pressure difference

The situation where there is subcooled refrigerant at the entrance to the capillary, whether caused by suction line heat exchange or otherwise, is discussed. A curve is presented to show that sub-cooling increases the refrigerant circulation; in other words a longer tube or one with smaller inside diameter is needed to maintain capacity balance with sub-cooled liquid at entrance.

Another graph is presented, labelled Capillary Design Chart for both F12 and F22. This shows a variation of friction factor with Reynolds number. A single line appears, marked Saturated Liquid, separating the sub-cooled liquid region from the Liquid and Vapour region. The curve is drawn from experimental data, to aid the designer when selecting a capillary tube.

Theory and Use of the Capillary Tube Expansion Device

M.M. Boletad and R.C. Jordan (5)

Part 1

This part deals with adiabatic conditions only. A graph is presented to show that the temperature remains constant while the pressure drop is linear to a point when the pressure equals the saturation pressure for that temperature. Beyond this point both temperature and pressure fall, both corresponding to the saturation conditions.

A graph is presented to show that the evaporator pressure has very little effect on the mass flow rates. Therefore the back pressure was excluded from further consideration. Another graph shows that a higher initial pressure increases the flow rate, as does the initial sub-cooling.

A graph is presented to show that if an oil separator is used, the flow rate falls by about 8 percent, i.e. the presence of oil increases the mass flow rate, contrary to the statement made by Stoebler (31).

Using the steady flow energy equation

$$h_0 + {}_0q_1 = h_1 + \frac{V_1^2}{2gJ}$$ (between Sections 0 and 1)

and the equation of continuity

$$\frac{W}{A} = \frac{V_1}{v_1} = \frac{V_2}{v_2} \quad \text{etc.}$$

Bolstad and Jordan have deduced the relation

$$dl = \frac{Tds - dq}{\frac{fv^2}{2gDJ}\left(\frac{W}{A}\right)^2}$$

where h = enthalpy

q = external heat transfer

V = velocity

g = acceleration due to gravity

J = mechanical equivalent of heat

W = mass flow rate

A = cross sectional area

v = specific volume

l = length of tube

D = inside diameter of tube

T = temperature

s = entropy

Prefix d = differential

subscript 0, 1, 2, denote different sections of the tube, whence the length of capillary tube can be

determined knowing or assuming the factors in the R.H.S. of the equation.

There is no experimental evidence offered to substantiate the theory. It is merely stated that flow process is isothermal practically but not theoretically.

Part II - Non-Adiabatic Flow

Published later, this paper presents a graph showing the pressure and temperature drop along the tube length. The "flash-off" point referred to by Swart (31) is renamed bubble point temperature". This graph shows the temperature falling, due to heat exchange with the suction gas, while in the earlier adiabatic case it was shown to be isothermal. In the length of the tube after the heat-exchange, there is a small rise in temperature from the atmosphere. Pressure readings along the tube were not taken, but the graph is drawn as a straight line, using the inlet pressure, and tangent to the temperature curve at bubble point. Beyond this point, as stated in the earlier paper, the liquid temperature and pressure are those corresponding to the saturation values. No pressure readings are shown. The only effect of heat exchange is to shift the bubble

point further downstream to nearer the tube exit. For the all-liquid flow, the liquid temperature has little effect on the pressure drop.

It is then argued that the rate of pressure drop and the tube length, for a given flow rate, are the same for both the adiabatic and non-adiabatic cases, as long as both have the same bubble point temperature. The conclusion is drawn that for a given flow rate and bubble point temperature only one tube length is possible, regardless of adiabatic or non-adiabatic conditions of all-liquid flow.

Graphs are presented to show the actual effect of the three main variables; the inlet pressure, length of tube and internal diameter of tube on the flow rates, as observed in the experiments.

Friction factors were obtained from the flow of water through the capillary tube, and it is stated that these are valid for any other liquid or gas also.

In the discussion it is stated that where inconsistent readings were obtained much of the trouble lay with the oil content of R12 circulating through the system, the quantity probably varying with time of operation and ambient temperature. The oil content

was one factor which was never under control even with an oil separator in use.

In discussing the limitations of the theoretical solution in the first part, it is stated that undue amounts of oil present are likely to change the refrigerant viscosity by a significant amount, which would alter the Reynolds number and friction factors. Laminar flow was observed, evidenced by a low friction factor, when the N_e number computed from the viscosity of the pure refrigerant indicated turbulent flow. The oil also depressed the boiling pressure and may have altered the flow condition in the two phase portion of the flow. The oil probably affects the surface tension also.

An example of the application of the theoretical equations is presented, and the heat exchange is analysed in another.

Pressure Drop with Change of Phase in a Capillary Tube

C.P. Marcy (22)

Marcy's paper gives a theoretical approach to the two-phase flow of refrigerants through capillary tubes.

The equation is presented

$$N = \frac{G^2 g D}{G^2} \int_{P_2}^{P_1} \frac{\rho}{f} \, dp$$

where N = length of tube, ft., needed to reduce pressure of saturated liquid to evaporator pressure.

G = weight rate of flow, lb/sec - ft^2

D = inside diameter of tube, ft.

g = acceleration due to gravity, ft/sec^2

ρ = density of fluid, lb/ft^3

f = friction factor

The function ρ/f is assumed continuous and single-valued, and graphical integration is carried out to obtain a solution.

When sub-cooled liquid is entering the capillary, it is assumed to flow adiabatically till the pressure reaches saturation, and the length of tube

required for this may be calculated seperately, using the same equation for liquid flow condition. For adiabatic flow conditions, the liquid flows isothermally up to the saturation point. The liquid density may, therefore, be assumed constant.

The equation is rearranged to solve for flow rate, under given length conditions.

Results are presented for F12 and SO_2, to show close agreement between the computed and test data.

No mention is made of any oil mixed with the refrigerant nor of any steps taken to eliminate the oil during these experiments.

Rating the Restrictor Tube

N.E. Hopkins (13)

The capillary tube is described here as the restrictor tube. The author feels that the word 'restrictor' is more descriptive of its behaviour in the control of refrigerant, as the tube does not function like a capillary in ordinary use.

Two series of graphs, described as rating curves, are presented to aid the designer in the selection of the tube to give the required performance. The consideration of sub-cooled refrigerant at capillary entrance, either due to suction heat exchange or otherwise, is included in the graphs, which deal with R12 and R22.

The series I rating curves give more accurate results than the series II curves, due to certain approximations in the latter. An example is that no provision is made for suction temperature other than 40°F. The Series II graphs are, however, easier to use, and the approximations do not have much effect on accuracy under actual conditions.

Calculated results compared closely with the test results, especially those given by Holstad and

Jordan (9), on the pressure and temperature drop along the tube, and the effect of bubble point temperature.

The friction factors are nearly 13% higher than those given by McAdams (23). Initially it was found that calculated restriction lengths, using McAdams friction factors, were 13% greater than actual lengths.

The Adiabatic Flow of Evaporating Fluids in Pipes

D.C. Lanning (19)

This investigation deals with evaporating water, and R12 flow through pipes. Interest was concentrated on determining the flow forms that occur.

While the flow of evaporating water adopts the form of separated or annular flow, R12 is likely to have frothing flow. The theoretical formulae deduced are based on this assumption. The experimental results presented agree closely with the predicted results.

The R12 mass flow rates are tabulated in the form $lb/sec\text{-}ft^2$. No justification is given for this. On the contrary, the mass flow does not vary directly as the square of the diameter of the tube, and so the flow rates on this basis are likely to differ widely for tubes of different bore.

Lanning has, however, used only one tube of 0.042 in. bore. Graphs are presented showing the fluid temperature along the tube for different mass flows.

Lanning has assumed thermal equilibrium, and in the discussion it is held that if equilibrium is

maintained at the tube outlet, there is no reason why it should not hold at other sections where the proportion of liquid present in the mixture is greater. It is claimed that the evidence is sufficiently strong to be accepted as conclusive.

Analysing the frothing characteristics of a fluid, the author considers that (a) a low value of surface tension coefficient is encouraging to froth formation, and (b) solutions are more susceptible to frothing than pure liquids.

Metastable Flow of Freon 12

P.F. Pasqua (25)

Stable equilibrium theory states that when the properties of a liquid are changed such that a liquid vapour phase boundary is crossed, vapour will appear.

Pasqua examined the mass flow of R12 through nozzles, orifices and short tubes. The experimental mass flow were always considerably in excess of the quantities predicted by the stable equilibrium theory. The conclusion was that the liquid R12 continues to flow as a liquid, perhaps in a super-cooled state, described by Pasqua as a "metastable" state.

Based on the experimental results, Pasqua presents equations to predict mass flow and determine critical pressures under metastable flow conditions.

Turbulent Flow of Air through Capillary Tubes

P.G. Smith (30)

A method of calculating the air flow rates through capillary tubes is presented. The equation

$$\text{Air flow} = K \frac{(P + 2.3)^{0.94} \, d^{2.6}}{l^{0.505}}$$

where K = a constant

P = inlet pressure

d = inside diameter of tube

l = length of tube

is deduced from experimental values, and is therefore valid within the experimental range, inlet pressures from 10 to 100 psi, air flow of 0.10 to 0.60 cfm, lengths from 2 to 20 feet, and inside tube diameters from 0.030 to 0.090 inches.

Evidently the equation is not valid for zero pressures and gives infinite flow for zero length (orifice).

The average value of the constant K is given as

K = 69.73 with standard deviation = 0.367

A table of experimental values of K is presented.

The Coefficient of Heat Transfer for Boiling Freon-12 in Horizontal Tubes

S. Pierre (26)

This paper deals with the heat transfer coefficient in evaporators. Several methods have been used to obtain a theoretical equation for the heat transfer, considering, complete evaporation, and incomplete evaporation. Comparison has also been made with available experimental results. The experimental results, however, are diversely varied, but no explanation is given for this.

The flow form prevailing inside the tubes is not discussed.

No conclusions have been drawn in this article.

This paper was brought to the author's notice by several sources, and a review is presented here to show that it has no bearing on the present problem, and because of the difficulty experienced in obtaining a copy of the publication.

Capillary Two Phase Flow

H.A. Mitchell (36)

Part I

Mitchell developed two formulae, one for non-condensible and non-volatile liquid, such as air and water flowing together, the second for two-phase flow of NH3. Slug flow was assumed, with reservations about its validity. Isothermal flow was assumed, because liquid is present.

The equation for NH3 is derived for the general case where both saturated liquid and gas are present at the capillary inlet. It is then modified for the particular application where only saturated liquid enters the capillary.

$$M = \sqrt{\frac{0.22 P_1^{1.333} \left[1 - e^{\frac{190}{0.333} P_1} \left[\left(\frac{P_2}{P_1}\right)^2 - 1\right]\right]}{\frac{2 f_m L}{D} + \log_e \frac{P_1}{P_2 e^{\frac{190}{0.333} P_1} \left[\left(\frac{P_2}{P_1}\right)^2 - 1\right]}}}$$

Friction factors used are calculated from

$$f_M = 1.5\, f_l\, (1 - \beta) + 1.5\, f_g\, \beta$$

where M = rate of fluid flow

p = absolute pressure

e = base of natural logarithms

f = friction factors

L = length of capillary tube

D = inside diameter

β = weight ratio of vapour to total fluid flowing at entrance

and Subscripts l = liquid

m = mixture

g = gas

1 = inlet

2 = outlet

The theoretical and experimental mass flow results presented agree closely. The highest inlet pressure given is 79.0 psig. The pressures, however, are not measured directly, they are the saturation pressures corresponding to the temperatures measured.

Part II

An equation is derived for application to HFC. The equation, however, is the same as that

developed for R12 if the relation Reader equation is re-defined as

$$f_m = K(f_l \frac{M_l}{M} + f_g \frac{M_g}{M})$$

where K = ratio between two-phase and single phase values, and other nomenclature regarding the same.

Here graphs on the flow of R12 and also on R22 are presented to show the agreement between the theoretical and experimental values. Charts are presented to simplify design calculations.

All data presented are, however, limited to saturated liquid, or saturated liquid and varying vapour quantities at entrance to capillaries. The question of sub-cooled liquid at entrance, the most common case in practice, is not dealt with.

All experiments were conducted using refrigerant from bottles, and are, therefore, oil-free.

Simple Selection Method for Capillaries Derived from Physical Flow Conditions

L. Cooper, C.K. Chu and V.R. Erickson (7)

Equations are presented to determine the length of a capillary for the desired flow. The values of the Fanno lines were calculated on a computer.

The basic relations used are

(1) The Law of Conservation of Mass

(2) Newton's Second Law of Motion

(3) The First Law of Thermodynamics

(4) The Second Law of Thermodynamics

Friction factors were also calculated; the equation

$$f = \frac{0.791}{R_e^{0.25}} \quad \text{where Reynolds number, } R_e = \frac{GD}{\mu}$$

where G = flow density lb/min-in^2

D = inside diameter

μ = viscosity

was used for liquid flows, and for two-phase flow account was taken of the vapour formed. Reynolds number was made a function of refrigerant quality.

$$R_e \text{ mixture} = R_e \text{ liquid} + \frac{x}{2}(R_e \text{ gas} - R_e \text{ liquid})$$

where X is the quality of vapour at the tube exit.

Actual flow measurements showed that calculated lengths were less than known lengths in every case. Tests on a glass capillary tube showed the visible liquid lengths to be greater than those calculated from the pressure drop. The authors conclude that there is a delay in vaporization; that is, thermal equilibrium is not maintained. Further downstream there is a rapid transition and equilibrium is restored.

The delay factors were calculated from actual data; the calculated delay was really a correction factor, incorporating all unaccounted-for variables. It was observed that delay

(1) increases with decreasing inside diameter

(2) increases with increasing length

(3) increases with increasing inlet pressure.

Capillary tube performance graphs are presented for R22, and it is stated that the results could be applicable to Refrigerant 12 and steam also.

Some Characteristics of Flow Pattern and Heat Transfer of Freon 12 Evaporating in Horizontal Tubes

P. Worsøe-Schmidt (37)

Although this paper does not directly concern flow in capillary tubes the flow patterns described by the author are of interest.

Worsøe-Schmidt studied the flow pattern in glass evaporator tubes, similar to copper tube evaporators on which heat transfer measurements were carried out, and describes changing flow patterns from separated flow with smooth interface to wavy or stratified and slug flows; annular and semi-annular flows were also observed in certain portions of the evaporator.

One part of oil per thousand refrigerant was described as oil-free refrigerant. Investigations were conducted with controlled quantities of oil varying from 1 to 15 percent. Even small quantities of oil allow a film to be formed around the tube, permitting annular flow. Although oil-rich film cannot be supposed to contribute significantly to the heat transfer, the active part of the tube wall is still

greater than it would be if no oil was present; hence an increase of heat transfer was obtained.

Graphs are presented to show the variation of heat transfer coefficient with different quantities of oil. The heat transfer coefficient increases with larger quantities of oil to a certain limit, in the earlier sections of the tube. Though the coefficients decrease in the latter sections of the tube, with increasing quantities of oil, there is a definite overall increase in heat transfer due to the presence of oil.

Chapter 5

DISCUSSION

5.1 Evaporating Flow

5.2 Metastability

5.3 Flow Form

5.4 Non-Evaporating Two-Phase Flow

5.5 The Solution

5.6 Viscosity

5.7 Density

5.8 Reynolds Number

5. INTRODUCTION

Among the numerous uses of the capillary tube, two are concerned in this report.

(1) The capillary tube may be used as an expansion device or restrictor in a refrigeration circuit, in place of an expansion valve, located between the condenser and the evaporator. Such capillary tubes are widely used in refrigeration systems, particularly in the smaller and hermetic units. The flow through this tube is primarily refrigerant in the liquid state, with small quantities of oil dissolved in the refrigerant. Throughout this report, the capillary tube used for this function is referred to as the "expansion capillary".

The expansion capillary is sometimes connected thermally to the compressor suction line; a length of the tube, at the condenser end, is soldered to the suction line, to act as a heat exchanger. Here the spent suction gases are used to further sub-cool the liquid refrigerant in the capillary tube, resulting in an increase in the coefficient of performance of the system. Virtually all published work available on capillary tubes with R12 and referred

to in this thesis is related to this application.

(2) The capillary tube may also be used as oil-return line from the oil separator to the compressor crankcase, as a substitute to the valve mechanisms currently in use. It is the possibility of this application of the capillary tube that is the primary concern of this report, and the tube when so used, is referred to as the "oil-return capillary". No published work is available relating to this possible application.

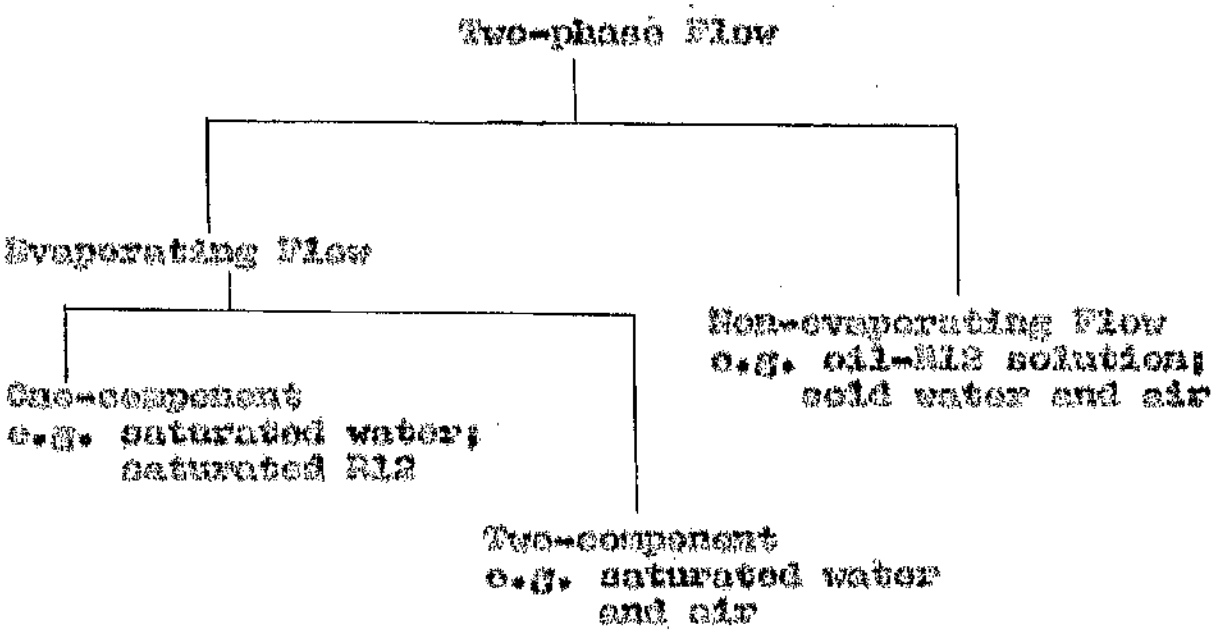
This report concerns itself with the possible use of the capillary tube as an oil-return device, the present problem can be stated as an investigation into the mass flow rates of various concentrations of oil-R12 solution through capillary tubes of various lengths and diameters, for varying conditions, and the possibility of predicting such mass flow rates.

The oil-R12 solution normally encountered in refrigeration systems contains a low pour point, high viscosity lubricating oil, the R12 vapour is dissolved in the oil. In the flow of such solutions through capillary tubes the possible reappearance of R12 vapour puts the problem into one category of two phase flow.

The complex subject of two-phase flow of fluids through tubes has interested engineers for many

ponse, due to the presence or application of the phenomenon in many situations.

Two-phase flow can be sub-divided, (1) Evaporating two-phase flow, and (2) Non-evaporating two-phase flow. Examples of two-phase evaporating flow are (a) saturated water and (b) saturated R12. The flow of saturated water and air is an example of two-component evaporating two-phase flow. The present study of oil-R12 solutions may be classified as two-component non-evaporating two-phase flow. The flow of cold water and air may also be considered two-component non-evaporating two-phase flow, with nearly constant ratio of gas to liquid.



5.1 Evaporating Flow

In evaporating two-phase flow, such as occurs in the expansion capillary, saturated or sub-cooled liquid enters the capillary, and during its flow through the tube, there is a drop in pressure, due mainly to friction. The pressure may be so reduced that the sub-cooled liquid becomes saturated, and vaporisation commences, assuming thermal equilibrium, with absorption of latent heat of evaporation from the surrounding medium. Pasqua (25) has defined stable thermal equilibrium: "When the properties of a liquid are changed, such that a liquid-vapour phase boundary is crossed, vapour will appear." This describes what might be expected in the flow of saturated liquid through a capillary tube. The presence of this vapour flowing simultaneously with the liquid would constitute the two phases. A vapour bubble may appear. This point has been described as "flash off" point by Swart (33), and "bubble point temperature" by Holstad and Jordan (5). The flow upstream of this point is liquid and therefore virtually isothermal, and downstream the flow is two-phase. If the inlet temperature and the pressure distribution along the tube are known the point where evaporation should commence could be located. For an adiabatic flow

the temperature of the fluid should fall at this point. Though the fluid temperature falls, there should be no change in the total enthalpy, if the frictional losses are neglected.

Bolstad and Jordan (5) presented graphs to show the pressure and temperature gradients along the expansion capillary tube for R12 flow. For adiabatic flow the graph shows a straight line pressure while the temperature remains constant until saturation is reached. In some applications the expansion capillary is soldered to the suction line of the compressor to allow the colder suction gases to further subcool the liquid refrigerant in the expansion capillary, resulting in an increase in the coefficient of performance of the refrigeration system. For such suction line heat exchange, the graphs show a temperature drop causing the liquid R12 to be further sub-cooled, while the pressure, as in the adiabatic case, drops in a straight line until saturation point. When the saturation point is reached, both the temperature and pressure fall corresponding to saturation conditions, irrespective of adiabatic or non-adiabatic liquid flow. Suction line heat exchange, at the condenser end of the capillary, merely moves the bubble point further downstream. These observations are

corroborated by Hopkins (13).

Lanning (19) in his investigation on the flow of R12 through a 0.042 in ID capillary tube also assumed that thermodynamic equilibrium was maintained, and that standard thermodynamic theory for compressible fluid flow was applicable. The limited range of experimental results appeared to substantiate the assumption.

3.2 Metastability

Cooper, Chu and Brinkan (7) observed during refrigerant flow through glass evaporator tubes, that the visible liquid lengths were greater than those calculated assuming thermal equilibrium. The conclusion from this work was that there is a delay in vaporization; that is, thermal equilibrium is not maintained. This phenomenon was also observed by Norsø-Schmidt (37) in glass-tube evaporators, where the length of liquid flow was greater than calculated on the assumption of thermal equilibrium. Pasqua reported that thermodynamic equilibrium is not maintained during the flow of R12 through expansion devices. Emerson (9), in his investigation on the flow of R12 through expansion devices has corroborated this, and has presented experimental data on different

shapes and sizes of nozzles, short tubes, and orifices. Psaras and Bureau conclude that the effect is due to a metastable state of supercooled liquid.

On this evidence it is possible that metastable conditions will occur also in the flow of R12 through expansion capillary tubes. Therefore, it would be difficult to locate the bubble point.

5.3 Flow Form

Previous investigations on the flow of R12 assumed frothing flow, but no evidence is advanced to support this. In the only attempt to study the flow form of evaporating R12 Lanning (19) described three possible forms of flow for evaporating fluids: (a) Annular flow, in which the liquid phase is confined to the wall of the pipe, the vapour flowing in an inner core. (b) Separated flow, when the phases occupy segmental areas of the tube cross-section, the vapour flowing above the liquid; the interface may be smooth, wavy or stratified, depending on the relative velocities. (c) Frothing flow, when the liquid surrounds the vapour in the form of small bubbles, interspaced; the liquid and vapour phases flow with a common velocity.

In annular and separated flows, the liquid and vapour phases have relative velocity between them.

From continuity, momentum and energy requirements, Lin[illegible] deduced a relation for "relative velocity factor", the value of which gives an indication of the flow form to be expected. From the experimental results Lin[illegible] concludes that frothing flow prevails in the flow of evaporating R12. He states that solutions, in general, are more susceptible to frothing flow than pure liquids, and that a low surface tension coefficient is encouraging to frothing flow.

5.4 Non-evaporating Two-phase Flow

Two-phase flow may also occur where vapour is formed without the vaporization of saturated liquid. The conditions governing this type of flow are different from evaporating two-phase flow, and may be classified as non-evaporating two-phase flow. The flow of oil-R12 solution in the oil-return capillary is in this category since there is no evaporation of either oil or R12 during flow. Any R12 vapour formed is by release from solution. There being no vaporization, no latent heat is absorbed. However, a heat quantity may be involved depending on whether the solution is

exothermic or endothermic, but the quantity of heat will be very small compared to the latent heat of evaporation. With oil solutions generally, it is unlikely that there is any heat of solution, involving oil and R12.

5.5 The Solution

A distinction must now be made as oil dissolves R12 vapour at a pressure and temperature where R12 does not normally exist in the liquid phase. While in solution, the R12 is not in the vapour state either, but free R12 would be in the vapour phase under similar conditions. R12 vapour is released from solution because of a lower saturation limit at a lower pressure or higher temperature. It is, therefore, established that there is no evaporation nor absorption of latent heat, when R12 is released as a vapour from the solution. Also, it is not known if R12 in solution may be considered to have the thermodynamic properties of a vapour or a liquid.

In a mechanical mixture, the two or more substances exist side by side, each retaining its individual physical, chemical and thermodynamic properties, and are separable by simple mechanical means. The mixtures are usually heterogeneous.

When two or more substances react chemically, the products are different from the reacting substances, and would have different properties, bearing no resemblance to the original substances.

Where two substances mix together, such as oil and H2O, without any resulting phase difference in the mixture, such solutions cannot be termed mechanical mixtures, nor do they react chemically. The solution is homogenous, and in certain cases may involve a heat of solution. There may also be ionization and formation of charged particles. The solvent and solute retain their identity, and chemical composition, but may undergo a structural rearrangement in the molecules, to form a bond between the two substances. This causes properties like specific weight to be altered, and it is not possible to predict the properties of solution from the properties of its components.

Oil is not a pure compound, but a mixture of compounds having varying boiling points, which undergo polymerisation and degradation with changes in temperature; this adds to the complexity of the problem.

Supersaturation in a solution may be compared

to super-cooling in a liquid, where vaporization can be delayed if heat is applied slowly, and there are no impurities present. Any disturbance or addition of any impurities restores thermal equilibrium and evaporation takes place. Likewise supersaturation may be attained in a solution, delaying the release of vapour from solution.

The process of dissolving H_2S in oil is not an instantaneous one; there will be a certain time required to obtain a saturated oil-H_2S solution. Similarly, there will be a time lag for the release of H_2S vapour from a saturated oil-H_2S solution, if the pressure is lowered or the temperature raised. The time required in the latter case is expected to be less than the time required to form a saturated solution. This time lag will depend on the rate of change of pressure or temperature, and must be a factor where the rate of change is rapid as in the present problem, where in some circumstances conditions permit the release of H_2S vapour from solution.

5.6 Viscosity

The viscosity of an oil is lowered by any H_2S dissolved in it. The lower viscosity of solutions

will result in lower frictional losses, and mass flows will be greater than in the individual flow of oil only, under the same conditions. The viscosity of oil-R12 solutions will depend on the concentration, in addition to the temperature of the solution.

The mass flow through the capillary tube will be affected by the degree of concentration.

5.7 Density

The density of oil will be increased by any R12 dissolved in it; the solution will be heavier than oil or R12 vapour individually. Since mass flow is a function of density, the increased density of the solution will result in higher mass flow.

Accurate information on the viscosity-temperature and density-temperature relations of oil-R12 solutions at the relevant conditions are not available, and attempts to obtain such information from the manufacturers of Capella D oil, The Regent Oil Co., Ltd., and the manufacturers of R12, E.I. DuPont de Nemours & Co. Inc., did not meet with any success.

5.0 Reynolds Number

Reynolds number for the flow of oil only through capillary tubes shows that laminar flow could be expected. The lower viscosity and higher density of oil-R12 solution, both contribute to increasing Reynolds number to a value which may lie in the transition range, further increasing the complexity of the problem.

While the complexity of the problem hinders a theoretical solution, it was considered a comprehensive series of results would be useful to the designer of the capillary tube as an oil-return line from the oil separator to the compressor crank-case and would reveal the magnitude of trends and effects which at present are matters of surmise.

Chapter 6

AIM and SCOPE

OF

THE EXPERIMENTS

6. AIM AND SCOPE OF THE INVESTIGATION

The purpose of the investigation was to examine the possibilities of using the capillary tube as an oil return line from the oil separator to the compressor crank-case, to provide a simple substitute for the valve mechanism commonly used.

The experiments were, therefore, designed to present comprehensive data on the flow of oil-R12 solutions through capillary tubes of various bores and lengths. Because of a dearth of published information on the properties of oil-refrigerant solutions, it was necessary to make certain assumptions in compiling the results. The manufacturers of the oil, The Regent Oil Co. Ltd., and the manufacturers of Freon 12, E.I. DuPont De Nemours & Co. Inc., were unable to supply the data needed on proportion at the relevant temperatures and pressures.

Comparison has been made between the individual theoretical and actual flows of oil and of R12.

Chapter 7

THE EXPERIMENTAL PLANT

7.1 Flow Measurement

7.2 Initial Design

7.3 The Need for a Change of Design

7.4 Final Design

7.5 The Refrigeration Circuit

7.6 The Capillary Tube

7.7 Pressure Taps and Nitrogen Calibration

7.8 Pressure Gauges

7.9 Thermocouples

7. THE EXPERIMENTAL PLANT

Experience proved that it was necessary to modify the normal refrigeration circuit to enable experiments to be conducted on the flow of oil-R12 solutions. The difficulties faced were new and unusual in the experimental field of refrigeration.

The two prime requirements were that the plant (1) should be capable of circulating large quantities of oil with refrigerant in solution, and to maintain the head and suction pressures for test periods, and (2) to be able to measure the quantities of oil and R12 accurately and without affecting the circuit.

7.1 Measurement of Mass Flows

The problem of mass flow measurement was two-fold. It was necessary to know the exact quantity of oil flowing through the capillary tube, and it was also essential to know the quantity of R12 dissolved in the oil thus flowing. The measurements had to be carried out without affecting the circuit charge quantity, so as not to disturb the equilibrium conditions of operation.

The methods considered were

(1) Rotameter

(2) Secondary evaporator

(3) Radioactive tracers

(4) Photo-electric Comparison with Standard Solutions

(5) Gravimetric Sampling

(1) Rotameter - The rotameter is used often in the measurement of fluid flows, and is reasonably accurate, when the physical properties of the fluid are known. In the present case it was necessary to measure the flow of oil-Rll solutions of varying unknown concentrations and at different pressures and temperatures. The physical properties of oil and Rll, individually, are known, but virtually nothing is known about the physical properties of oil-Rll solutions, apart from the elementary fact that concentration of oil-Rll solutions have to be known before any attempt can be made to predict the properties of oil-Rll solutions. Knowing the concentrations in use was a prime requirement. It proved impracticable, therefore, to use the rotameter for flow measurement in this situation.

(2) The Secondary Evaporator was used by some previous investigators, to measure the flow of refrigerant. In this equipment the evaporator coils are placed in an

insulated vessel. The cooling load is provided by a secondary refrigerant, heated by electric heaters, and condensed by the evaporator coils. The power input to the heaters is measured. The evaporation surface and heat transfer coefficient of the evaporator coils is known. Using the thermodynamic properties of the secondary and the primary refrigerants it is possible to measure to a reasonable accuracy the rate of flow of the primary refrigerant. To reduce errors due to heat loss from the secondary refrigerant to the atmosphere, the temperature of the secondary refrigerant is maintained near the ambient temperature, the pressure being controlled by the heat input to the heaters.

This method makes the assumption that the primary refrigerant is free from lubricating oil. This is not achieved even with efficient oil separators. The accuracy of measurement in the present situation would be reduced, since the thermodynamic properties of oil individually or oil-ill solutions are unknown.

(3) Radioactive tracers are being used to trace the path of flow by measuring the intensity of radioactivity. To serve the purpose of measuring mass flows in the present problem, it would require the isotope to have

an appreciably long half life, to reduce errors in quantity measurement. This would involve the risk to the operators of undue exposure to radioactivity. Another factor is that oil and H2O are completely miscible. Thus any tracer soluble in oil would also be likely to be soluble in H2O, making it impossible to obtain the correct concentrations of the oil-H2O solutions. Radioactive tracers are also known to cause wax formation and separation in the oil. It was, therefore, decided that radioactive tracers would not serve the present need.

(4) Photo-Electric Comparison with Standard Solutions - A method had been developed at the Royal College of Science & Technology, Glasgow (34), for determining the oil content of boiler feed water. Standard solutions of known oil content dissolved in ether were prepared, and the test samples compared by means of a Photo-Electric Fluorescence Comparator, the accuracy of the comparison was very high, up to 0.5 parts per million. The oil was insoluble in water, and thus test samples were made by dissolving the oil in ether. In the present problem, where oil and H2O are miscible, any solvent for oil is also likely to be a solvent for H2O, to a greater or lesser degree. It would be difficult

to extract oil from oil-R12 solution for the preparation of test samples for comparison with standard oil solutions.

(5) Gravimetric Sampling - This appeared to be a feasible approach. Although undesirable, a small quantity could be drawn off as a sample, and analysed gravimetrically. This would involve considerable skill and very accurate weighing, apart from the long time required for each test. The main objection to this method is the delay before knowing the oil-R12 proportion in the solution.

7.2 The Initial Design

A new experimental method was sought, to enable rapid and accurate measurements of mass flow of oil-R12 solutions through capillary tubes, and to evaluate the concentration of the solution.

The problem of mass flow measurement was solved by conducting intermittent tests on a by-pass to the main refrigeration circuit. This permitted large quantities of oil-R12 solution to be circulated through the capillary under test without interfering with the main refrigeration circuit.

A unique feature of the design was that it was no longer necessary to establish steady conditions of operation in the main refrigeration circuit. Steady conditions of operation of a refrigeration system depend on a large number of inter-dependent variables, evaporator ambient temperature, cooling load, motor supply voltage, condenser water temperature and pressure, compressor discharge and suction pressure, charge content in the system, and others. This made the process of obtaining steady conditions of operation a tedious and time-consuming one, a fact well-known to those who have experimented in this field. Conducting these tests on a by-pass made possible the complete control of test conditions, and steady conditions were readily obtained and easily maintained independent of the main circuit. The refrigerant compressor was used to maintain the required head and suction pressures and also to transfer the oil-RIS solutions from one receiver to another through pressure difference.

This has been a major development from any earlier experimental technique, and for the first time allowed the accurate control of mass flow of oil-RIS solutions of known concentrations to be handled under predetermined conditions.

(i) The Mixing Receiver was a vessel of approximately 0.3 cu.ft. internal volume, where the oil could be maintained at a constant temperature, pressurised by N_2 vapour, by-passed from the main circuit. The N_2 vapour was circulated through the mass of oil to ensure a homogeneous solution, and also helped to maintain the entire solution at a uniform temperature. The receiver was heated by external electrical heating coils. Fibre-glass insulation was used to reduce loss of heat to the atmosphere. An internal cooling coil was added later. The level of the solution was maintained above the entrance to the capillary - the liquid seal preventing any N_2 vapour from entering the tube. An external nylon tube indicated the level of oil inside. A thermo-couple was introduced into the vessel through a flange to measure the temperature of the solution, and a pressure gauge was connected to the receiver. The pressure could be raised by controlling the out-flow of N_2 vapour.

(ii) The Oil Receiver. The capillary tube exit was connected to the oil receiver - a calibrated glass vessel of about 1403 ml internal volume, with metal ends for the pipe connections, calibrated and able to withstand the experimental pressure range. The different densities and concentrations of the solutions

at different pressures and temperatures were accounted for in the volume measurements. The liquid settled at the bottom of the receiver and any R12 vapour released from the solution was led through a constant pressure valve at the top to the R12 receiver [(112) - described below]. External electrical heating coils were used to help the release of the refrigerant vapour due to the lower pressure, and to stabilize the remaining solution. A thermo-couple placed inside the receiver measured the temperature of the solution. A connection at the bottom allowed the oil to be drained away after each test.

The pressure in the oil receiver was maintained constant by a Danfoss PV valve, which controlled the outflow of R12 vapour. This allowed the capillary tube to discharge to a constant back pressure.

(112) The R12 Receiver was a vessel of about 0.80 cu. ft. internal volume. A low vacuum was obtained therein, before each test, with the help of the compressor. A combined vacuum and pressure gauge connected to the receiver indicated the internal pressure. The temperature was measured by a thermo-couple placed inside the receiver, through a flange. Having the final pressure and temperature, and using

the thermodynamic properties of superheated refrigerant vapour, the quantity of R12 vapour released was calculated.

Initially the volume of the receiver was determined from the weight of water, at known temperature, required to fill the receiver.

The compressor was used to build up and maintain the head pressure in the mixing receivers. The compressor was also used to create the necessary vacuum in the R12 receiver before each test. The oil from the oil receiver, after measurements were made for each test, was returned to the oil store and pre-heater by pressure difference created by the compressor. Thus the compressor was used as a valuable accessory for a number of functions.

A preliminary series of tests was carried out on the arrangement. It was observed that due to the intermittent nature of the tests, the capillary tube was heated by the warm fluids flowing through the tube, for an initial period of about 40 to 70 seconds. This transient condition prevented steady flow for that initial period and therefore introduced an element of error in the measurements. Steady conditions could only be obtained after the tube had warmed up to

the temperature of the solution. This was indicated by a noticeable change in the pressure drop along the capillary tube. The slope of the pressure drop curve was flattened with the attaining of steady conditions.

7.3 The Need for a Change of Design

This necessitated certain modifications in the design to remove the effect of initial transient conditions. It was decided to let the solution flow through the capillary tube, discharging into a separate receiver, for about 2 minutes, and then switching the exit into the calibrated oil receiver for a measured length of time. This had to be done without interrupting the flow through the tube.

7.4 The Final Design

(i) The Mixing Receiver - certain modifications were made to improve the mixing of the solution, and a sight glass was added to enable visual inspection of the entrance to the capillary and the condition in the mixing receiver. The internal cooling coil which initially carried evaporating refrigerant, in parallel with the evaporator, was replaced with a cooling coil to carry cold water. This was found to be more

effective.

(ii) The design of the oil receiver remained unchanged; the calibration was satisfactory, and the strength of the glass adequate to withstand the required pressures. Slight rearrangement of the heating coils was made to obtain more uniform heating.

(iii) The N2 Receivers - At the lower temperature tests, the capacity of the N2 receiver was not balanced with the oil receiver, and this made it necessary to increase the N2 receiver capacity. Two receivers, approximately the same size as the first one, were added in parallel to increase the N2 capacity. This prevented restraining the duration of a test, due to lack of adequate N2 capacity. The parallel arrangement of the three N2 receivers enabled the use of one or more receivers, as conditions dictated.

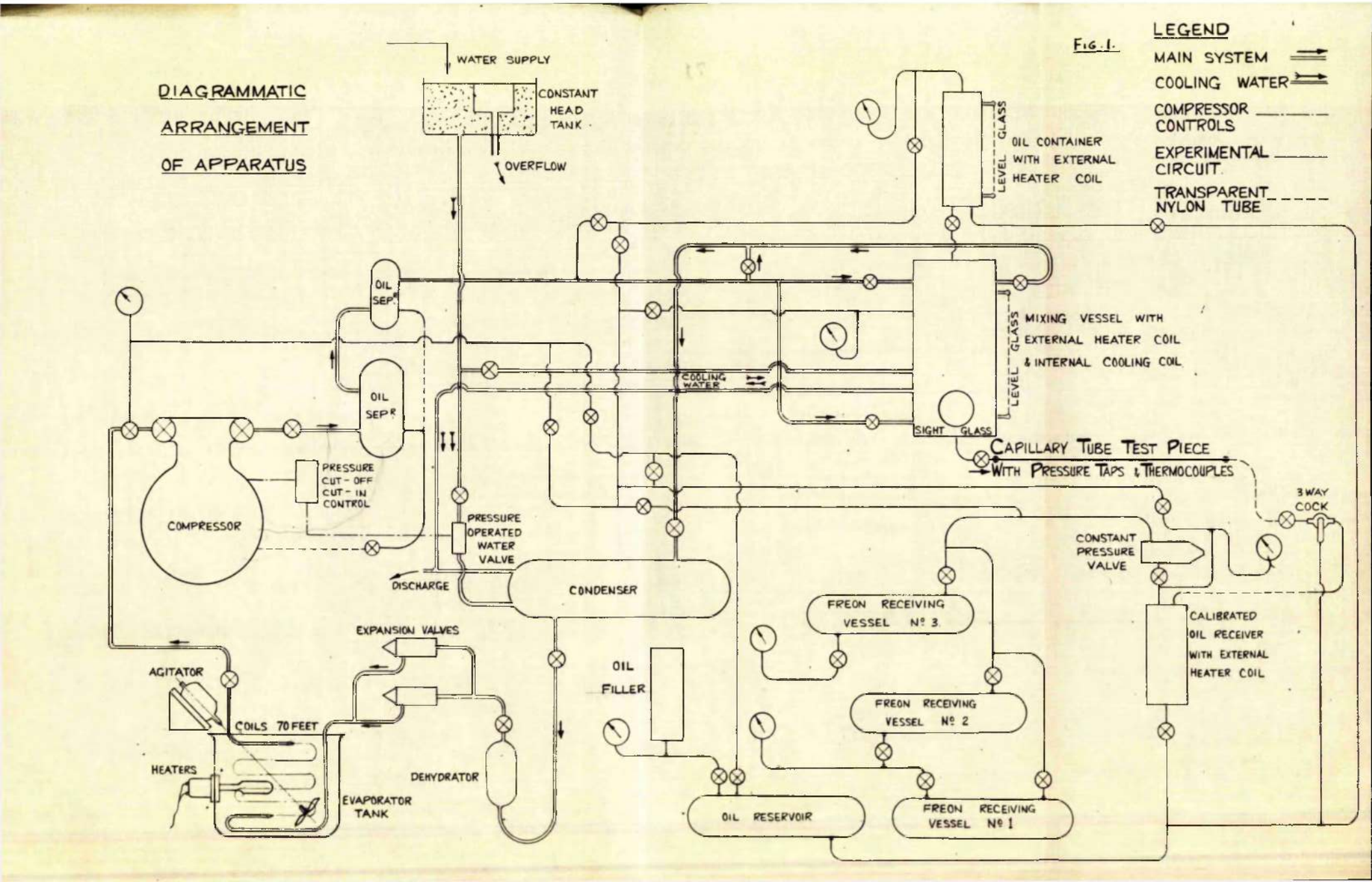
(iv) The Oil Reservoir - This was an addition to the original design. During the initial run of a test, the capillary exit drained into this oil reservoir, when transient flow conditions prevailed in the tube. The tube exit was switched over to the oil receiver, [(ii) above], after approximately two minutes without interrupting the flow. The upper

space in the oil reservoir was connected to the compressor suction; all the R12 vapour coming out of solution was taken away into the compressor, and the oil settled at the bottom of the reservoir. This helped to maintain the low pressure required at the capillary exit. The pressure in this receiver could also be raised through a connection to the compressor discharge, to allow the oil to be transferred to the oil pre-heater.

(v) The Oil Pre-heater. - This receiver had an approximate capacity of 0.1 cu.ft. and was placed directly above the mixing receiver, [(i) above], and was used primarily to introduce pre-warmed oil-R12 solution into the mixing receiver. It was connected to the discharge from the compressor, so that it was maintained at very nearly the high pressure maintained in the mixing receiver. This receiver could also be connected to the compressor suction to lower the pressure, when needed to draw up oil from the oil reservoir. A transparent nylon tube indicated the level of the oil in the receiver.

(vi) The 3-Way Cock - This was placed at the exit of the capillary tube, one outlet leading to the oil reservoir [(iv) above] and the other leading to the

Fig. 1.


DIAGRAMMATIC ARRANGEMENT OF APPARATUS
Fig. 1.
LEGEND
MAIN SYSTEM
COOLING WATER
COMPRESSOR CONTROLS
EXPERIMENTAL CIRCUIT
TRANSPARENT NYLON TUBE
WATER SUPPLY
CONSTANT HEAD TANK
OVERFLOW
OIL CONTAINER WITH EXTERNAL HEATER COIL
LEVEL GLASS
OIL SEP.R
OIL SEP.R
MIXING VESSEL WITH EXTERNAL HEATER COIL & INTERNAL COOLING COIL
COOLING WATER
SIGHT GLASS
CAPILLARY TUBE TEST PIECE WITH PRESSURE TAPS & THERMOCOUPLES
3 WAY COCK
PRESSURE CUT-OFF CUT-IN CONTROL
COMPRESSOR
PRESSURE OPERATED WATER VALVE
DISCHARGE
CONDENSER
CONSTANT PRESSURE VALVE
FREON RECEIVING VESSEL No 3
CALIBRATED OIL RECEIVER WITH EXTERNAL HEATER COIL
EXPANSION VALVES
AGITATOR
OIL FILLER
COILS 70 FEET
FREON RECEIVING VESSEL No 2
HEATERS
DEHYDRATOR
EVAPORATOR TANK
OIL RESERVOIR
FREON RECEIVING VESSEL No 1

oil receiver [(ii) above]. Only one outlet was in use at any one time. At the start of a run the cock arrangement allowed the fluid to flow into the oil reservoir. After about 2 minutes duration the cock was turned through 90° to allow the fluid to flow into the oil receiver, without interruption of the flow through the capillary tube. The test duration was timed from the moment the oil receiver was in circuit to the moment when the flow was cut off from the oil receiver.

(vii) The Oil Filler - This was an open receiver for the purpose of introducing oil into the system. The receiver when filled could drain into the oil reservoir, through a valve, aided by suction maintained by the compressor. Care was taken not to allow any air into the system when introducing the oil.

7.5 The Refrigeration Circuit

The refrigeration circuit comprised the usual components, compressor, oil separator, condenser, dryer, expansion device and evaporator. A schematic diagram is given in Fig. 2.

(i) The compressor was of the LL open-type, model

W150 Fil, loaned by Messrs. L. Stone & Co. Ltd., Glasgow. The compressor was driven by a 3-phase, 1.5 b.h.p. Brook electric motor, through a V-belt drive. An adjustable high-pressure cut-out was provided to stop the motor, and thereby the compressor, in the event of excessive pressure. The motor was also provided with a current overload cut-out.

(ii) Oil Separator - Two oil separators, the Danfoss 007 and 008 were used in series. This was not to prevent oil going into the circuit, where there was plenty of oil in any case, but to ensure adequate oil return to the compressor crank-case, to prevent it from running dry. The return line from the oil separator to the compressor crank-case was provided with a nylon tube connection for visual inspection of the flow of oil.

(iii) The condenser was water-cooled. Water was supplied from a constant-head tank through a pressure operated inlet valve. The compressor discharge pressure could be maintained by adjusting the David-Scott pressure-operated water inlet valve.

(iv) Dehydrator - A David-Scott Silica-Gel drier and strainer, D.E.35, was used between the condenser

and expansion valves to remove moisture or sediment that may have been in the circuit. The Silica-Gel element was rechargeable when required.

(v) Expansion Devices - Danfoss, AVV type, automatic expansion valves were provided, operating to maintain a constant evaporator pressure, and thereby a nearly constant compressor suction pressure. Two valves were used in parallel to provide the required capacity.

(vi) The evaporator was made up of approximately 80 ft. of $\frac{3}{4}$ in copper pipe, coiled and placed in a tank containing water. The tank capacity was 3 cubic feet. The cooling load was provided by 3 electric immersion heaters of capacity 1.5, 2 and 3 kW. A stirrer, driven by a small geared electric motor, was used to prevent the formation of ice on the surface of the evaporator coils.

The oil separators were placed just after the compressor discharge. The discharge, besides being connected to the condenser, was also connected in parallel to the oil mixing receiver, the oil pre-heater, the oil reservoir, and the oil measuring receiver.

The compressor suction, apart from drawing all vapour from the evaporator, was also arranged to draw from the oil reservoir, the oil pre-heater, the

oil mixing receiver, the B12 measuring receivers, and the oil measuring receiver. Suitable valves were provided to isolate or connect different sections as needed.

7.6 The Capillary Tube

Tests were carried out in two series. Four tubes were tested in the preliminary series, and five in the final series, comprising three internal diameters, and two lengths.

	Internal Diameter	Length
(a) Preliminary Series		
(1)	0.054 inches	3 feet
(2)	0.054 "	6 "
(3)	0.042 "	3 "
(4)	0.042 "	6 "
(b) Final Series		
(1)	0.036 "	3 "
(2)	0.054 "	3 "
(3)	0.054 "	6 "
(4)	0.042 "	3 "
(5)	0.042 "	6 "

The samples used were seamless copper tubes produced by the extrusion process, to a tolerance of

± 0.001 in. on the inside diameter, annealed, washed and cleaned to remove grease and dirt, tested for restriction, and their ends sealed before leaving the factory.

Pressure taps were provided in the tubes to observe the pressure drop along the tube. The drilling of pressure taps was a difficult process, and is described in a later section. In the preliminary series of tests, the 3 ft. length tubes were provided with 5 pressure taps at 6 inch intervals, and the 6 ft. length tubes were provided with 5 pressure taps at 12 inch intervals. From a study of the pressure drop along the tube in these tests, it was concluded that a closer study was needed of the pressure drop near the exit of the tube.

Consequently, in the final series of tests, the 3 ft. length tubes were provided with 6 pressure taps at 1 inch, 3, 6, 12, 18 and 24 inches from the exit of the tube. The 6 ft. length tubes were provided with 8 pressure taps, at 1 inch, 3, 6, 12, 18, 30, 42 and 54 inches from the exit of the tube. This permitted a closer study of the pressure drop near the exit of the tube, where pressure drop is fairly rapid compared to the upstream length of the tube.

Thermocouples were soldered to the surface of the tube to record the temperature drop along the tube. It was impracticable to introduce thermocouples into the tube to measure the temperature of the fluid directly, without seriously affecting the flow. It was therefore decided that a measurement of the surface temperature of the tube would have to suffice. It was estimated by Linning (19) that the surface temperature of the tube is less, due to loss of heat to the surrounding atmosphere. With adequate insulation, it was estimated that the difference in temperature would not be more than 0.2^{0}F.

The thermocouples were placed midway between two pressure taps, to avoid the cold spots that might occur due to the additional mass at the pressure tap. The thermocouples were coiled along the downstream length of the tube to reduce conduction in the thermocouple wires. The thermocouples, their calibration and methods of soldering to the tube are described in a later section, 7.9. The positions of the thermocouples were:-

Tube No. 1 0.036 in ID 3 ft length	Thermocouples placed at 2, 4½, 9½, 14 7/8, 21½, 29½, 34½ in from tube exit
Tube No. 2 0.056 in ID 3 ft length	Thermocouples placed at 2, 4½, 9½, 14½, 21, 29, 34½ in from tube exit
Tube No. 3 0.056 in ID 6 ft length	Thermocouples placed at 2, 4½, 8½, 14 7/8, 24, 37½, 49, 60½, 70½ in from tube exit
Tube No. 4 0.043 in ID 3 ft length	Thermocouples placed at 2, 4½, 8 7/8, 14 7/8, 21, 27½, 34½ in from tube exit
Tube No. 5 0.043 in ID 6 ft length	Thermocouples placed at 2, 4½, 8½, 15, 25, 36 7/8, 48½, 60 7/8, 70 in from tube exit.

7.7 Pressure Taps and Nitrogen Calibration

The small diameter of the tubes makes it difficult to introduce pressure taps without seriously affecting the flow. Several methods were tried without success. Only the method adopted successfully is described below.

The tubes were first cut to the required length, and tested for pressure drop with nitrogen gas. This a test with nitrogen after completion of the pressure taps, revealed the effect of the pressure tap holes on the flow. The arrangement used for the nitrogen test is given in Fig 2. The holes were then drilled very carefully with a drill of approximately

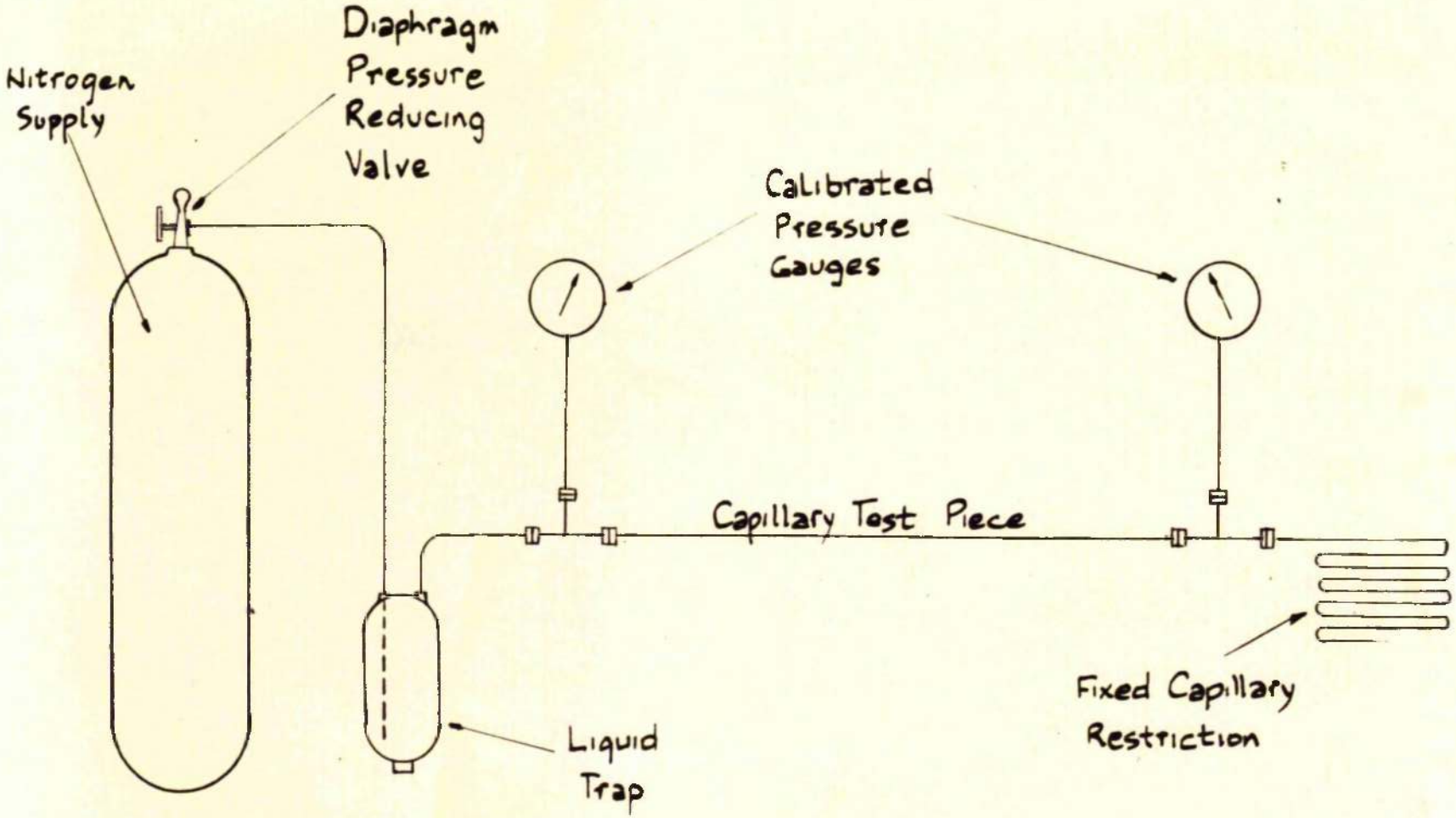


FIG. 2.

ARRANGEMENT OF APPARATUS FOR TESTING NITROGEN PRESSURE DROP ALONG CAPILLARY TUBE.

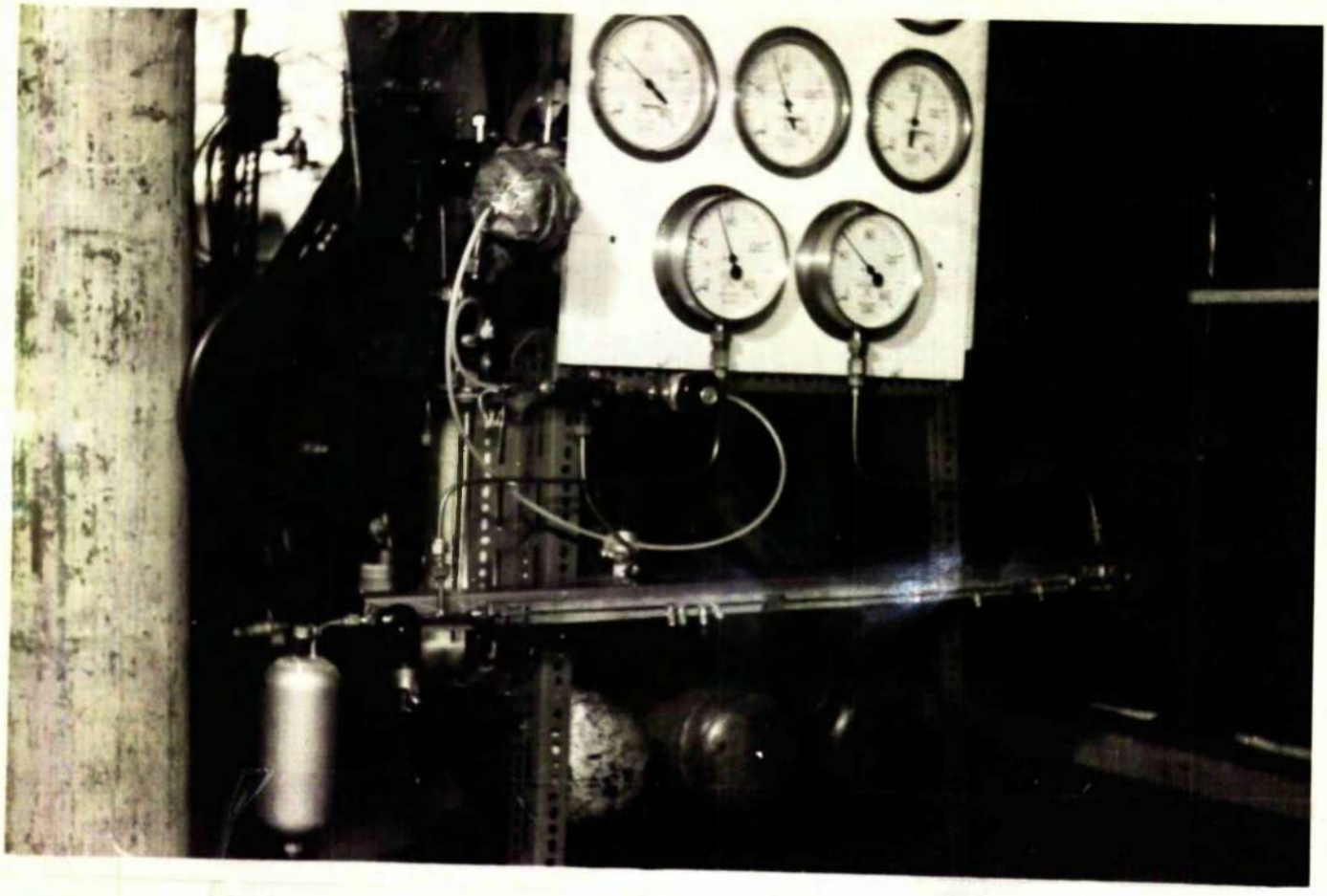


Fig. 3

Photograph showing the Nitrogen calibration arrangement in use.

0.013 in. diameter. Very small feeds were used when drilling to reduce the end burrs. However, it is not possible to avoid all burrs. Steel piano wire passed through the length of the capillary tube acted as a satisfactory reamer. The wire used was smaller than the bore of the tube, 0.035 in diameter wire for the tube of 0.036 in ID; 0.040 in diameter for the 0.042 in ID tube; and 0.052 in diameter wire for 0.054 in ID tube. After the holes had been drilled the connections were silver-soldered. Special gauge connections had to be designed. To reduce the chances of heat loss, the least amount of material was used, but it had to be strong enough to hold the pressure and make the connection possible. A detailed drawing of the gauge connection is given in fig. 4.

The tube was then tested for nitrogen pressure drop, as before. The two pressure drops were compared and if these were reasonably close, it was assumed that the pressure tap holes did not appreciably affect the flow conditions. The nitrogen calibration curves are presented in fig. 12-16.

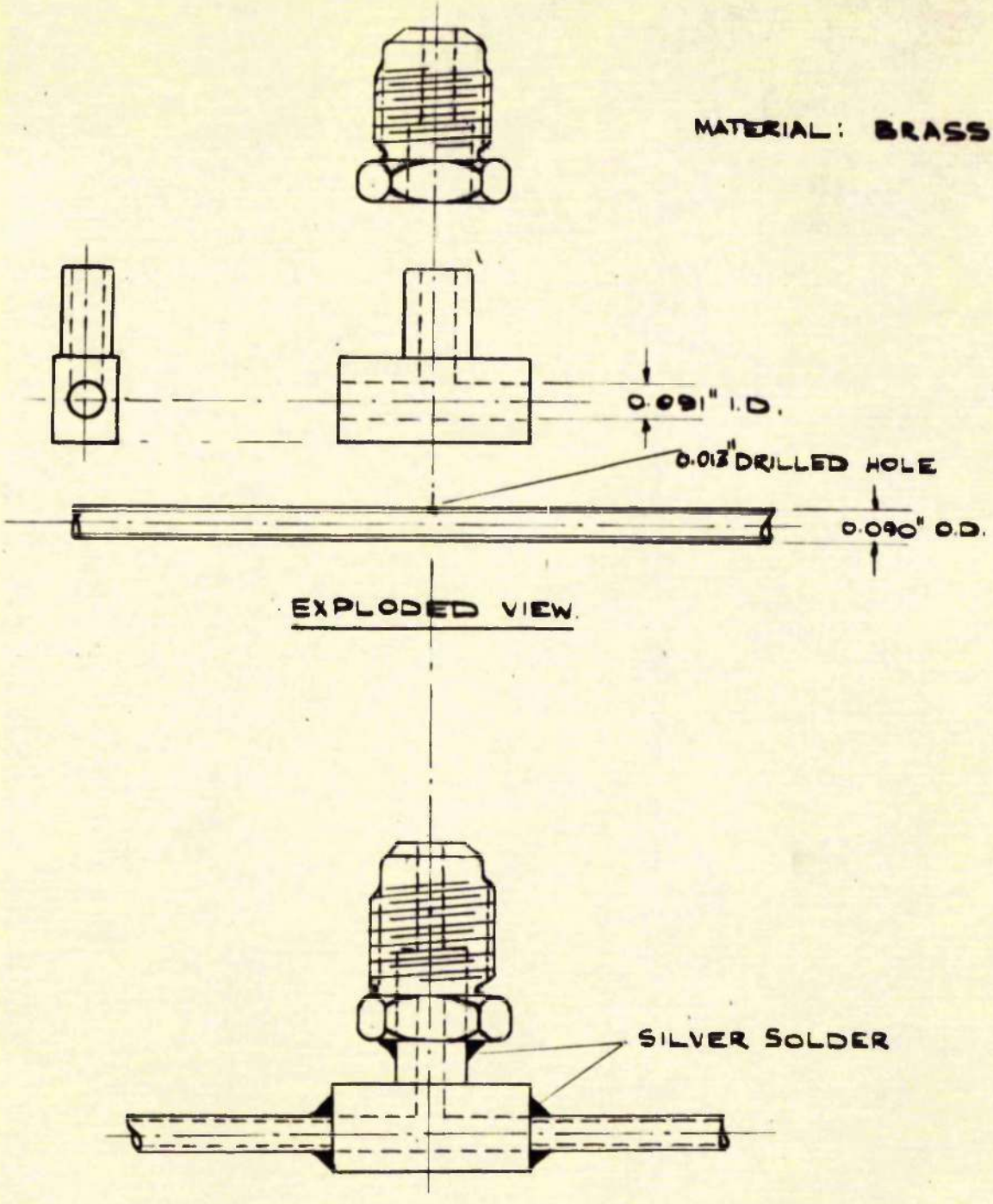


Fig. 4.

DETAIL OF PRESSURE TAP ON CAPILLARY TUBE

NOT TO SCALE

7.8 Pressure Gauges

Direct reading Bourdon-tube pressure gauges were used for the pressure measurements of the receivers and the pressure taps. For the R12 receivers also, Bourdon tube pressure gauges were used, but with a combined pressure and vacuum calibration. The gauges were manufactured by Messrs. C. & A. Stewart, Ltd. Glasgow.

All the gauges were calibrated, before being installed on the plant, and also at intervals, by a Dead-Weight Pressure Gauge Tester. The vacuum gauges, which were graduated to read inches of mercury, were calibrated against a column of mercury.

All gauge readings were corrected from their individual correction graphs, before being used in any calculation. All pressures reported, in tables and graphs, whether stated in absolute pressures or in gauge pressures, are the corrected pressures; the absolute being obtained by adding the barometric atmospheric pressure of the day to the corrected gauge pressures.

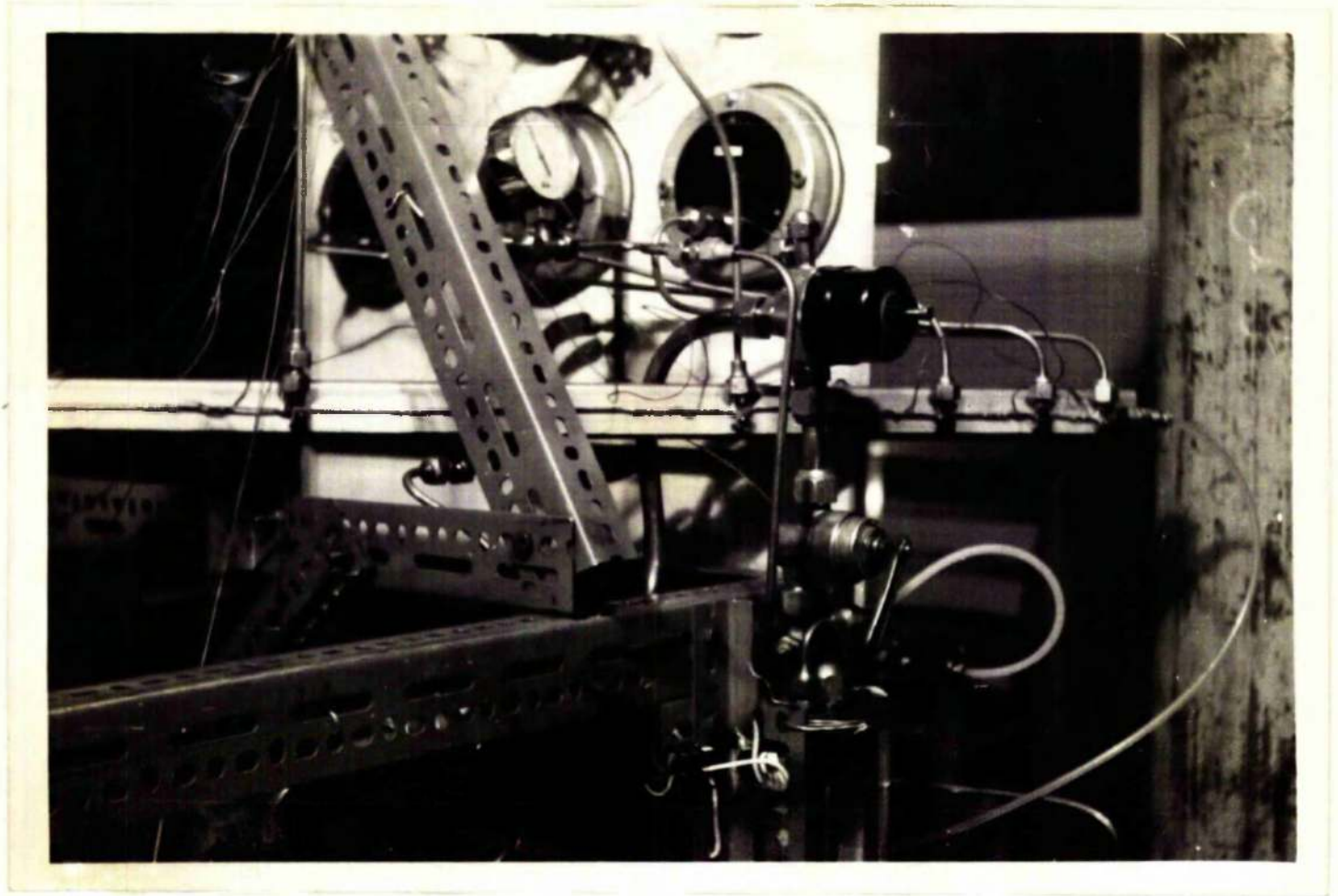


Fig. 5
Photograph showing the sample capillary tube in position. The pressure tap connections and the thermocouples soldered to the tube may also be seen.

7.9 Thermocouples

The thermocouples used on the capillary tube had to be fine because of the small size of the tube. The smallest available wire was 0.0095 in diameter, of chromel-constantan. The fineness of the wire reduced heat conduction losses, but these wires had a short life. In the final series of tests copper-constantan wires 0.007$ in diameter were used.

The thermocouple and the Cambridge potentiometer were calibrated jointly against a standard thermometer in a water bath maintained at various constant temperatures by a Circotherm II C.T. unit. The arrangement is shown in Fig. 7.

A multi-station selector switch was used to connect the various thermocouples to the cold junction and the potentiometer; the cold junction being maintained at 32°F, checked with a standard thermometer. Only the copper wire was coupled through the multi-station selector switch. All the constantan wires were soldered together at one point. This prevented stray e.m.f. being generated at the selector switch junctions due to any temperature difference between the copper and constantan wires. The copper wires were soldered to the selector switch where all

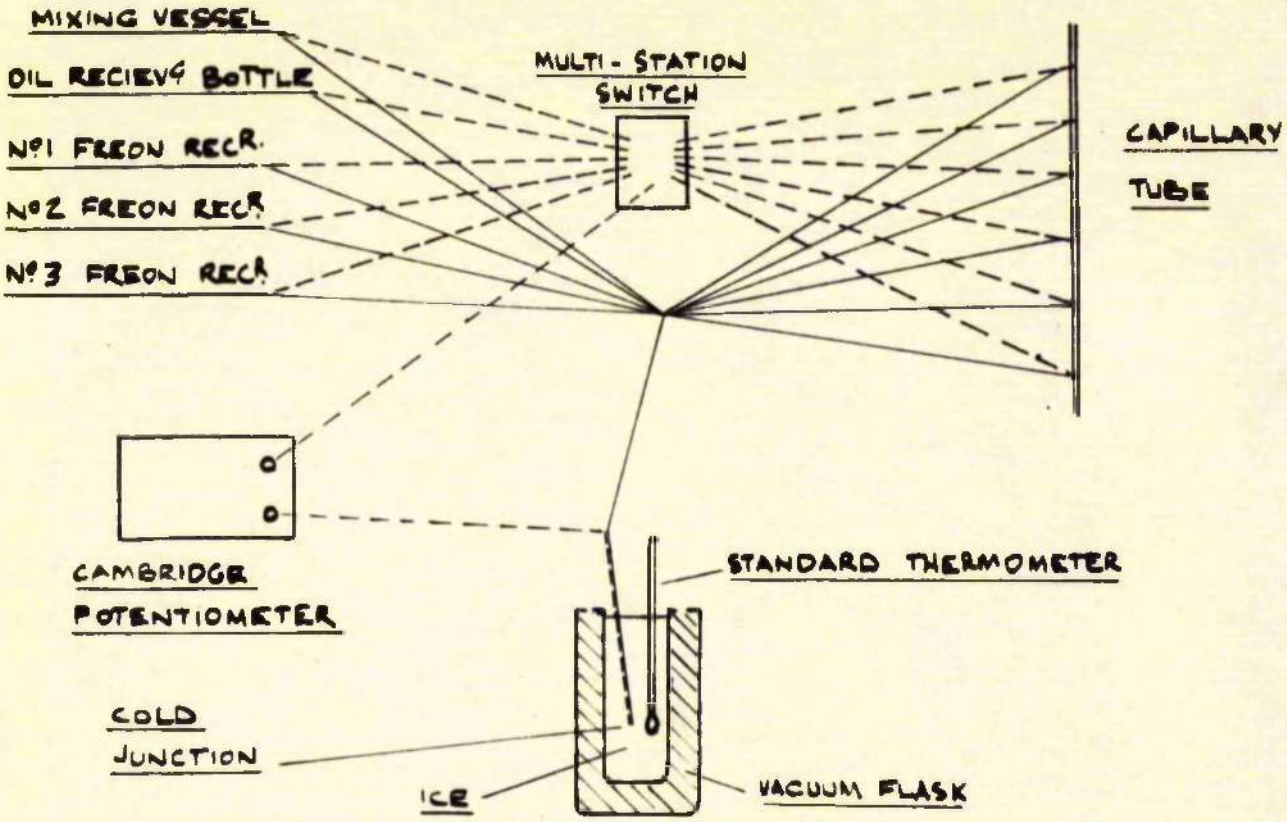


FIG. 6. THERMOCOUPLE WIRING DIAGRAM

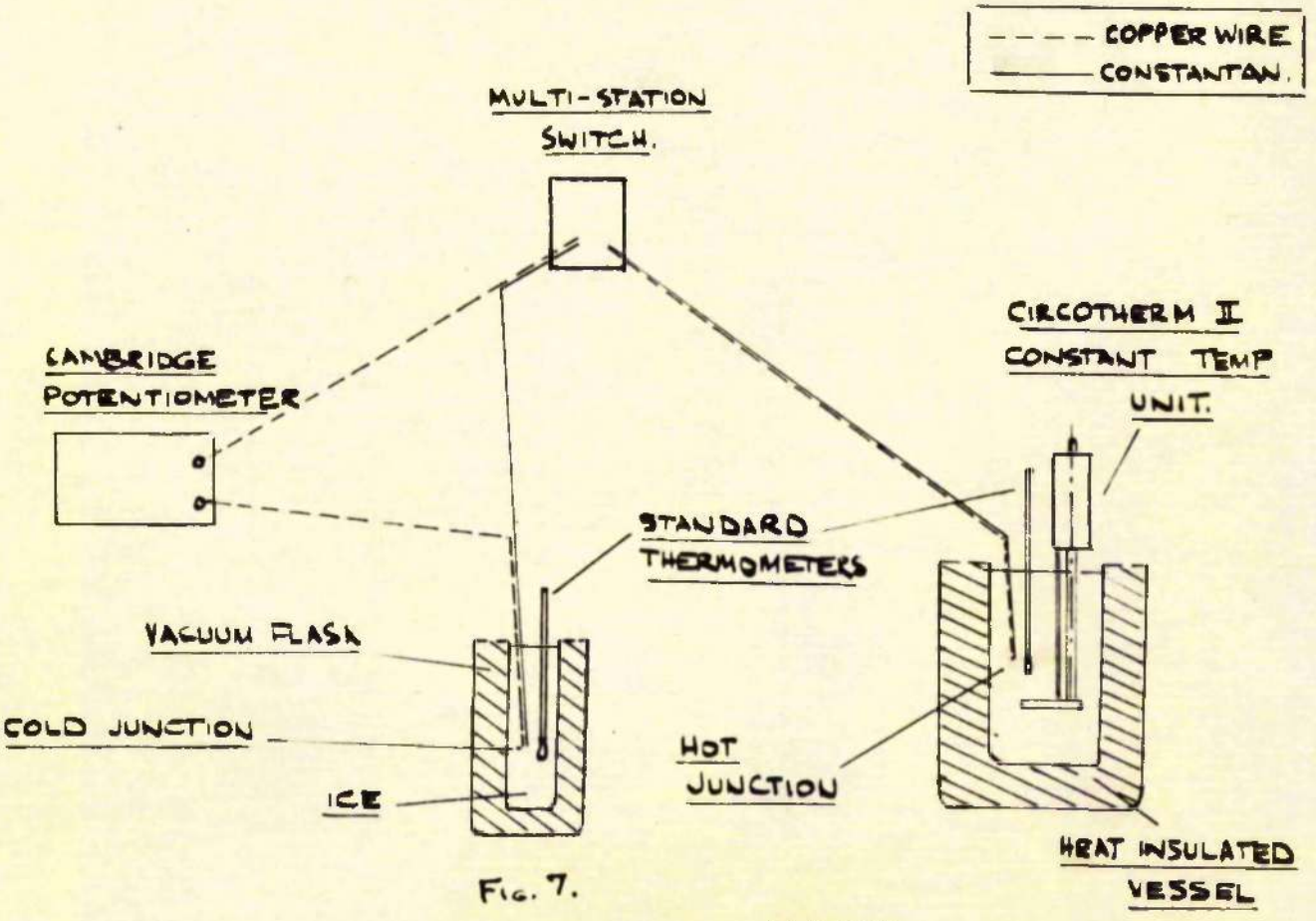


FIG. 7.

THERMOCOUPLE CALIBRATION DIAGRAM

contacts were of copper also and therefore any difference of temperature would not cause an e.m.f. to be generated. The wiring diagram is shown in fig. 6.

On the capillary tubes the thermocouples were soldered midway between two pressure taps, as described above, in section 7.6 the Capillary Tube.

Thermocouples were also used for measuring the temperature of the fluids inside the oil mixing receiver, the glass oil receiver, and the H2 receivers. In the case of the oil mixing receiver and the H2 receivers a pair of thermocouple wires were introduced inside each receiver through a flange connection. The wires were placed between a double joint to eliminate any chances of leakage. In the glass oil receiver the thermocouple wires were introduced into the vessel at the top collar connection, between a double joint.

After the parts were assembled, the plant was tested for leaks; nitrogen, in a suitable proportion, being added to the H2 to boost the pressure. When all leaks were sealed, the system was evacuated, using the compressor and an external vacuum pump. The plant was then re-charged with an adequate quantity of H2.

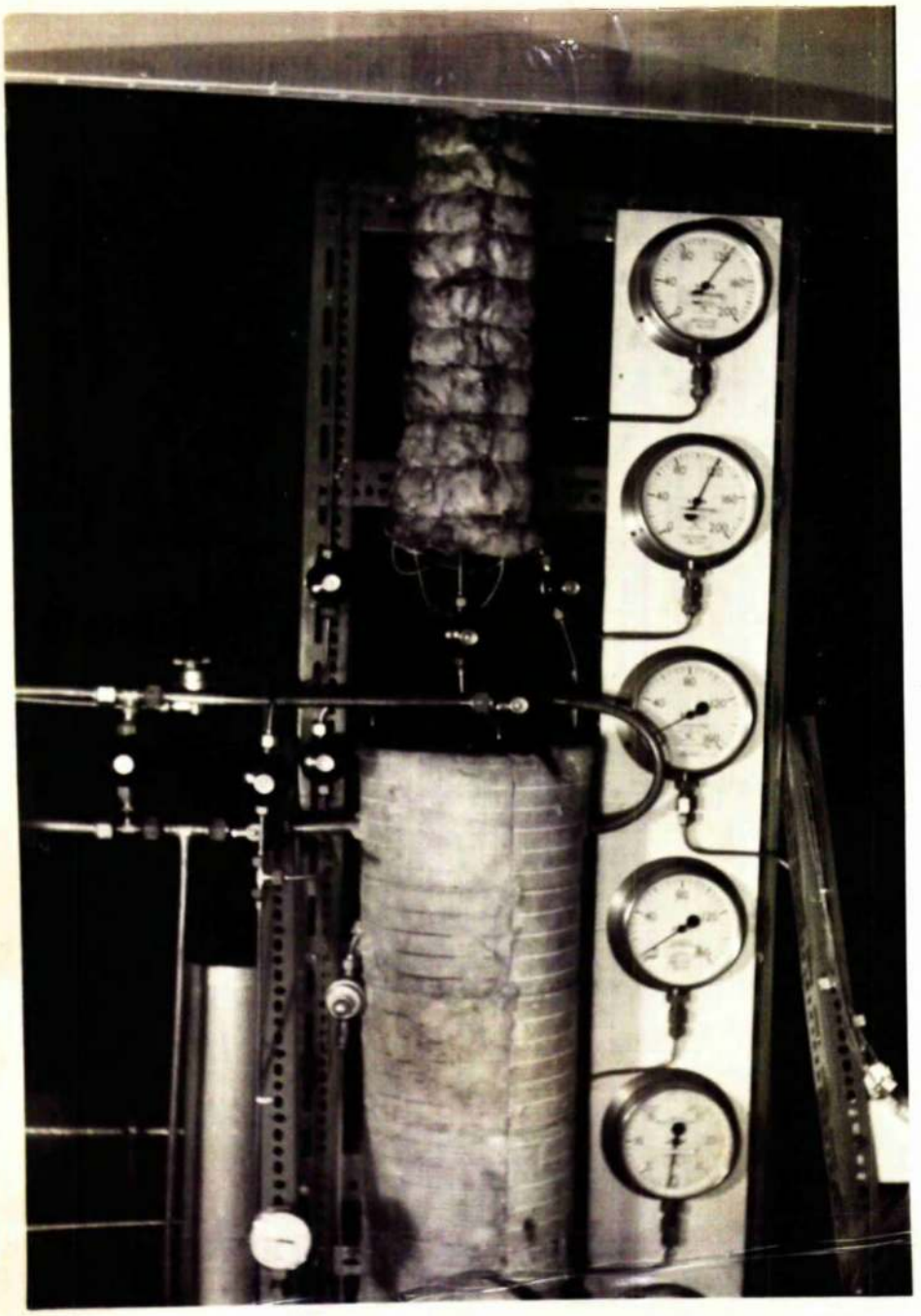


Fig. 8

A photograph of the upper portion of the experimental plant showing the H.P. mixing receiver, the oil pre-heater, the operational gauges, and some of the numerous valves.

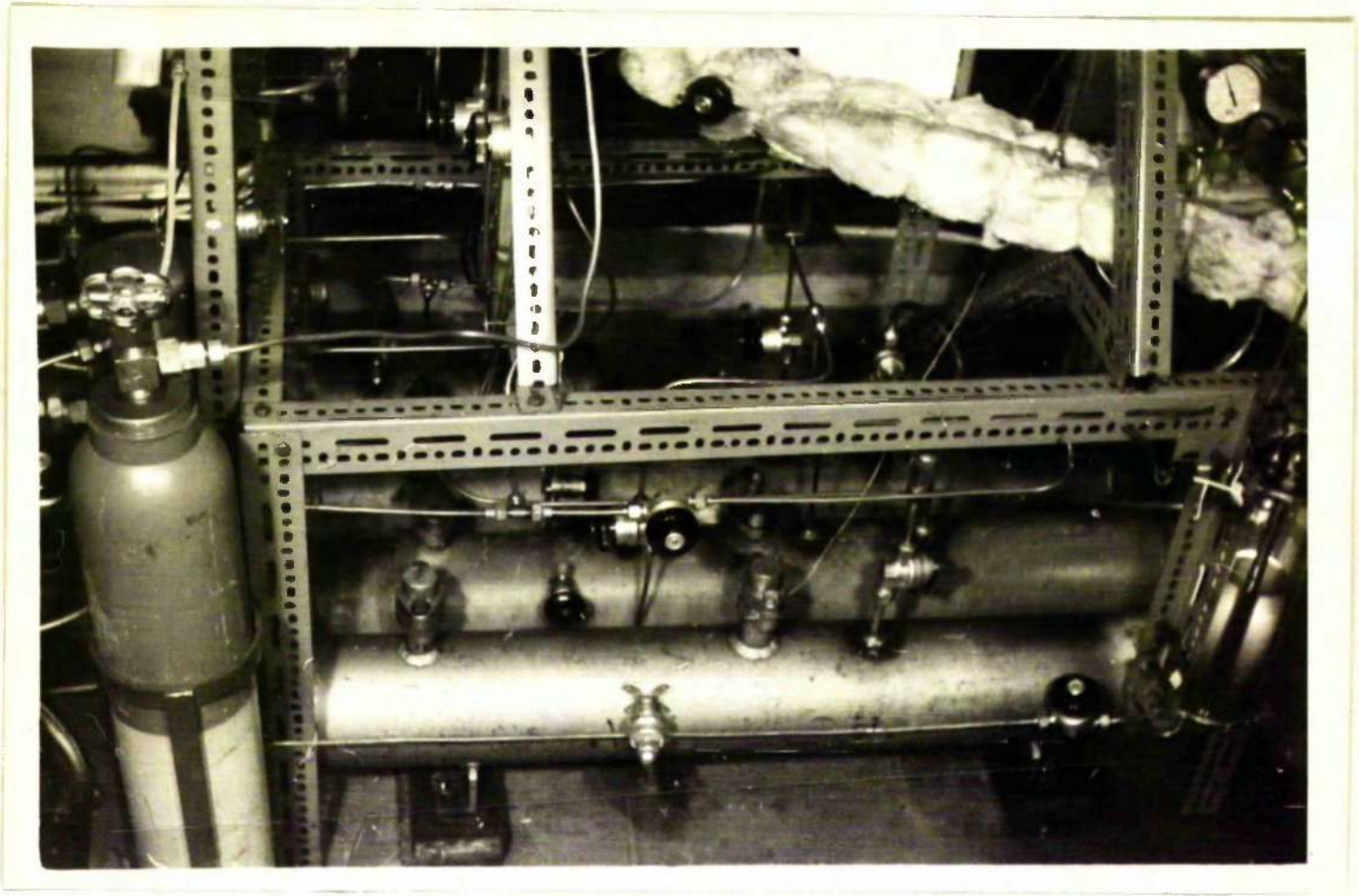


Fig. 9

A photograph of the lower portion of the experimental plant, showing the three R12 receivers at the bottom, the glass oil-receiver at the lower right hand corner. The sample capillary, lagged, can be seen at the upper right hand corner.

Chapter 8

[illegible]

6. EXPERIMENTAL PROCEDURE

Over 500 tests were conducted with oil-R12 solutions, and nearly 100 calibration tests using nitrogen. This large number of tests was made possible by the new techniques developed.

Oil was introduced into the system at the oil reservoir, through the oil filler. The oil was then transferred to the oil pre-heater and subsequently to the oil mixing receiver. The transfers were obtained by creating pressure differences between the receivers utilizing the compressor and the operation of the appropriate valves.

When sufficient oil had been introduced into the oil mixing receiver, it was heated by means of the external electrical heating coils. The pressure was raised and maintained by the compressor; the refrigerant passing through the oil helped to obtain a homogeneous solution at a uniform temperature. The receiver was then maintained at the temperature desired with the help of the heaters and the cooling coil as needed; the desired pressure was maintained with the help of the compressor.

Before starting the tests it was necessary to allow sufficient time for a saturated solution to be obtained at the desired pressure and temperature. The way of ascertaining whether a solution was saturated was to isolate the receiver by shutting the appropriate valves, and if the pressure did not fall while the temperature was maintained the solution was saturated. If a solution was not saturated there was a rapid fall in pressure as Rl3 vapour went into solution.

Tests were conducted only on saturated solutions, so it was possible to estimate the oil-Rl3 ratio from solubility charts. Measurements were made of the end products, of the oil saturated with Rl3 at the lower pressure and temperature in the oil receiver, and the separated Rl3. The Rl3 still dissolved in the oil is accounted for in calculating the quantities. The measured oil-Rl3 ratios have been compared with those given by the solubility charts.

Initially the oil receiver contained a small quantity of oil-Rl3 solution, and the remaining volume was filled with Rl3 vapour. Account had to be taken of this Rl3 quantity when the total quantities were calculated. Initial measurements were taken of the

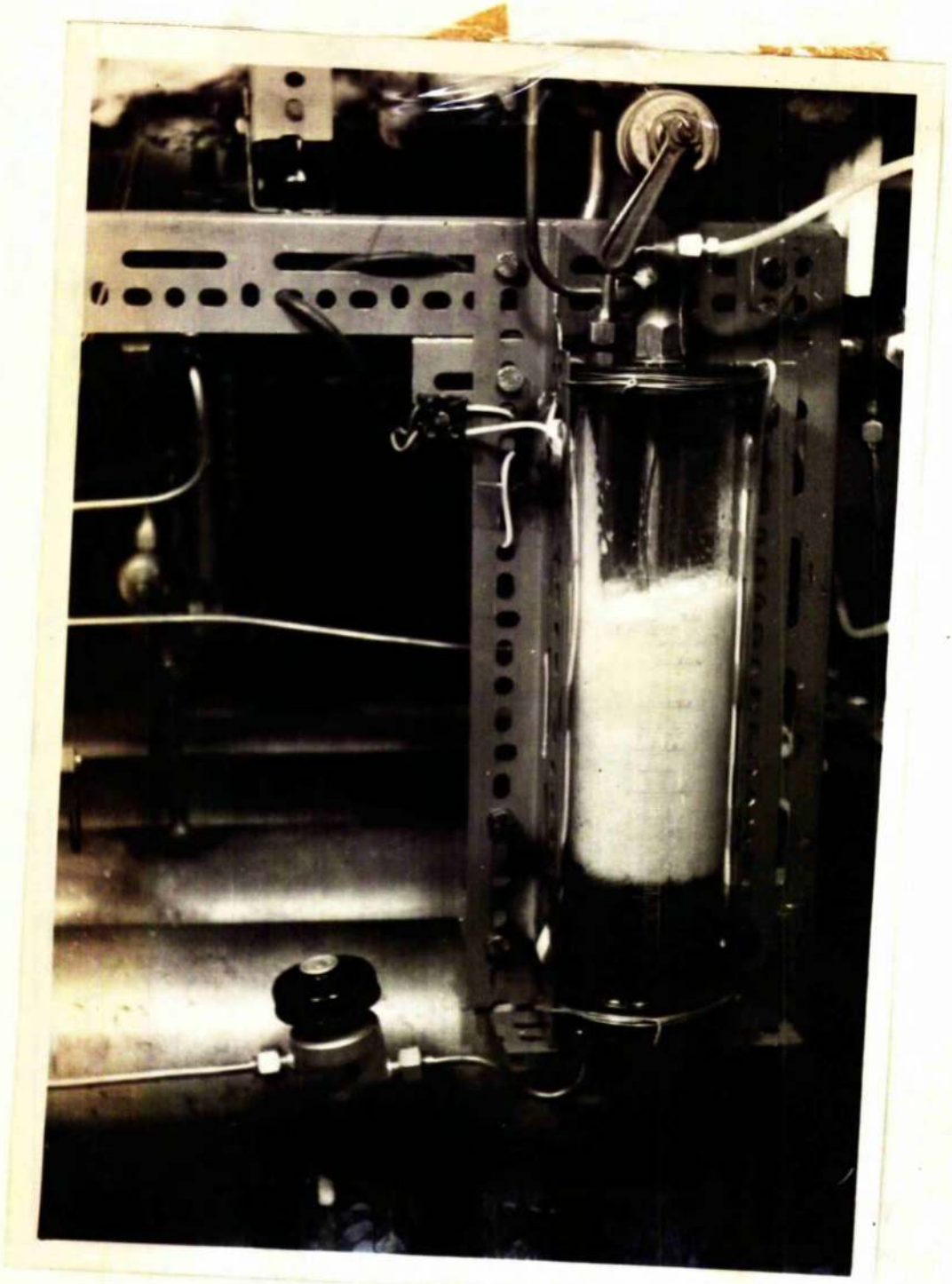


Fig. 10

Photograph showing the formation of froth in the upper layers of the oil-R12 solution in the receiver, during a test. The froth is caused by the release of R12 vapour from the solution due to a reduced pressure (lower solubility). Measurements are made when the froth subsides; the solution having reached equilibrium conditions at the reduced pressure and temperature.

volume of the oil solution (the total volume of the receiver less the volume of oil solution gave the volume of NH3 vapour), the pressure and temperature. Final measurements were also taken of these quantities after each test. The liquid remaining at the bottom of the receiver at any time was a saturated solution of oil and NH3, for that particular pressure and temperature, being subject to NH3 vapour pressure. Solubility curves were used to determine the percentage of oil and NH3. The total quantity of NH3, as vapour and in solution, and the quantity of oil, under the initial conditions were subtracted from the quantities under the final conditions, to obtain the actual quantities collected in the oil receiver for the particular test.

It was observed during a test that there was formation of froth in the top layers of the solution in the oil receiver: NH3 in the process of being released from solution at the reduced pressure. Such a froth formation may be seen in the photograph in Fig. 10, on the page facing this. The froth had to subside before any measurements could be made; the time required varied from about 4 to 12 minutes. Following the measurements, the oil was transferred to the oil reservoir, in preparation for the next test.

The R12 receivers were evacuated by the compressor before every test. The vacuum pressure and the temperature were noted. After the test was completed, the pressure and temperature were again recorded. The initial and final quantities of R12 vapour were calculated using the thermodynamic properties of superheated vapour. The difference gave the quantity of R12 collected in the R12 receivers for that particular test. This quantity of R12 was added to the R12 collected in the oil receiver, to obtain the total for that test.

When the temperature and pressure had been maintained constant for a reasonable period, a test was started. The valve at the exit of the capillary was opened, with the 3-way cock set for the solution to flow into the oil reservoir. This arrangement was continued for about 2 minutes, to allow the colder capillary tube to warm up to the temperature of the solution. To avoid errors due to this unsteady condition, the solution is diverted to flow into the oil reservoir instead of flowing into the measuring receivers. It had been estimated during the preliminary series of tests that the unsteady conditions prevailed for about 40 to 70 seconds. It was, therefore, considered that 2 minutes was adequate

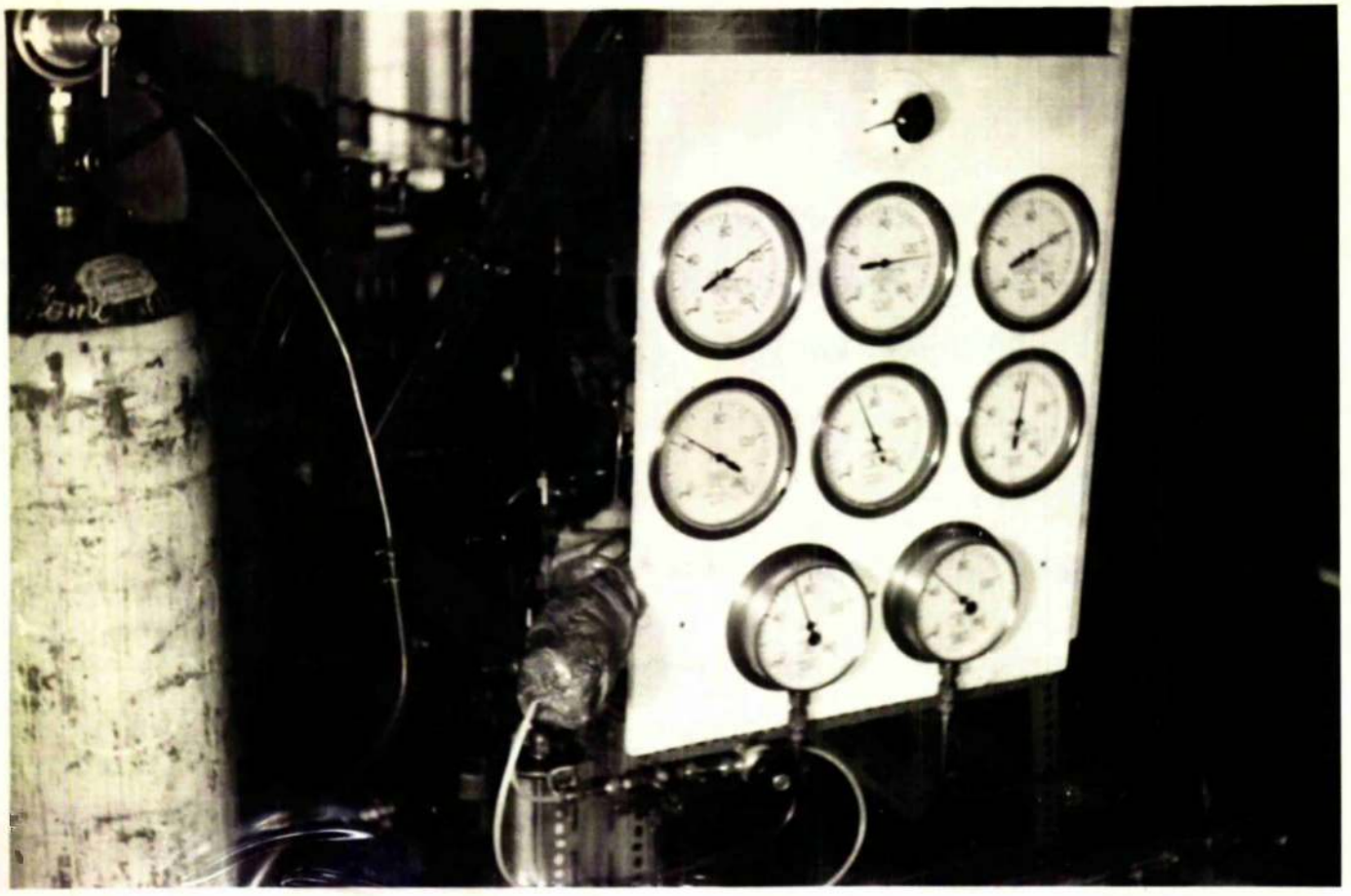


Fig. 11

Photograph showing the readings, during a test, on the pressure gauges connected to the pressure taps on thecapillary tube. The multi-station selector switch for the thermocouples may be seen at the top of the panel board.

time to obtain steady conditions. When the 3-way cock was switched over to allow the solution to flow into the measuring receivers; the start to end of the flow into the measuring receivers was timed with a stop-watch.

The solution poured down to the bottom of the oil receiver. A connection at the top allowed the "freed" NH_3 vapour to flow to the NH_3 receivers through a constant pressure valve, which was used to maintain a constant pressure in the oil receiver, thereby providing a constant back-pressure to the capillary.

While the solution was flowing through the tube the temperature on its surface was recorded by means of the thermocouples soldered to the tube. Pressure readings were also noted from the pressure gauges connected to the pressure taps.

Care and deftness was needed in conducting a test. Many valves had to be opened and shut during the test period, but some experience soon made the operator familiar with the process.

Chapter 9

DETERMINATION OF RESULTS

9.1 Nitrogen Calibration

9.2 Pressure and Temperature Distribution

9.3 Variation of Mass Flow Rate

9. EXPERIMENTAL RESULTS

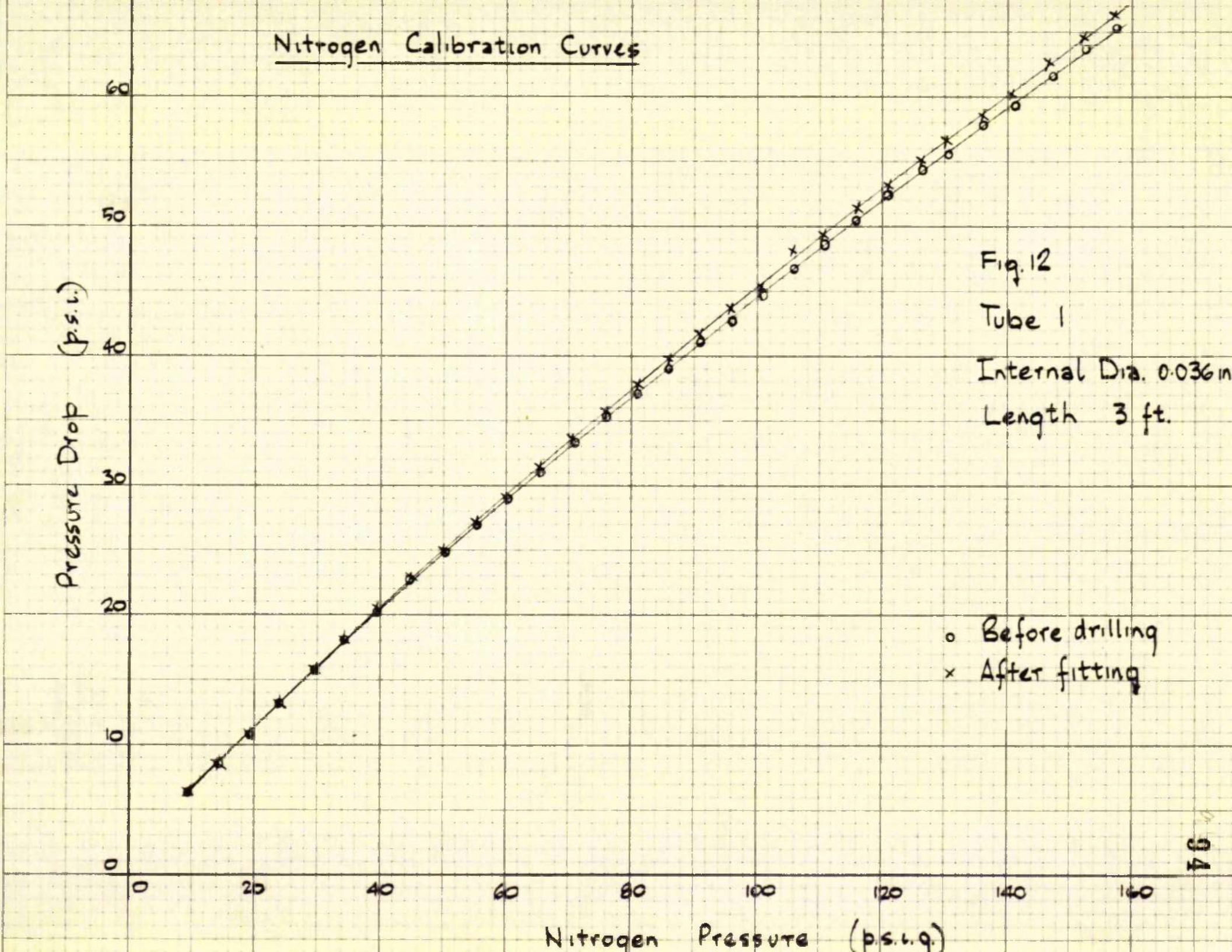
9.1 Nitrogen Calibration Curves

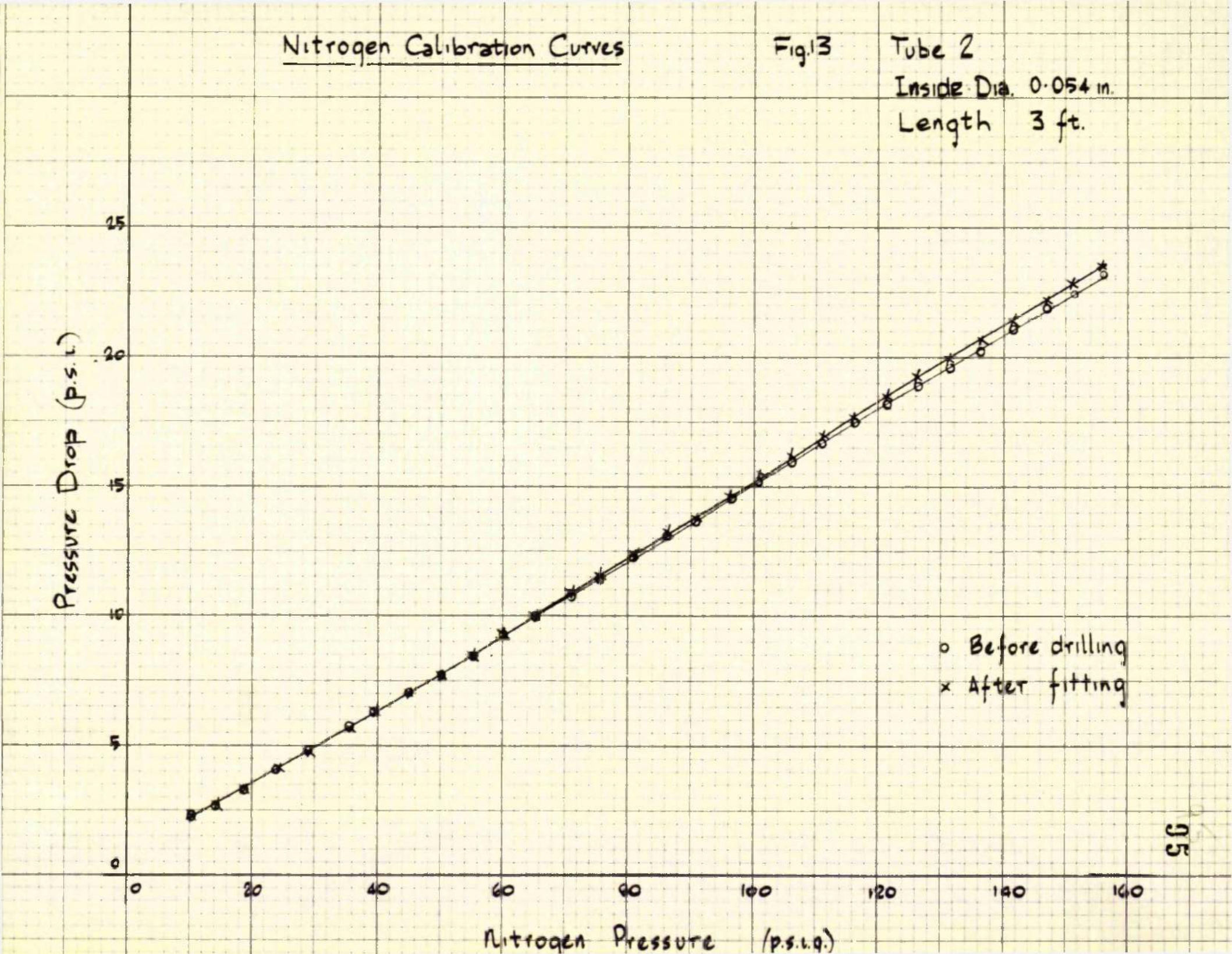
Each capillary tube, cut to the required length, was tested for nitrogen pressure drop. Readings were repeated several times to ensure consistency.

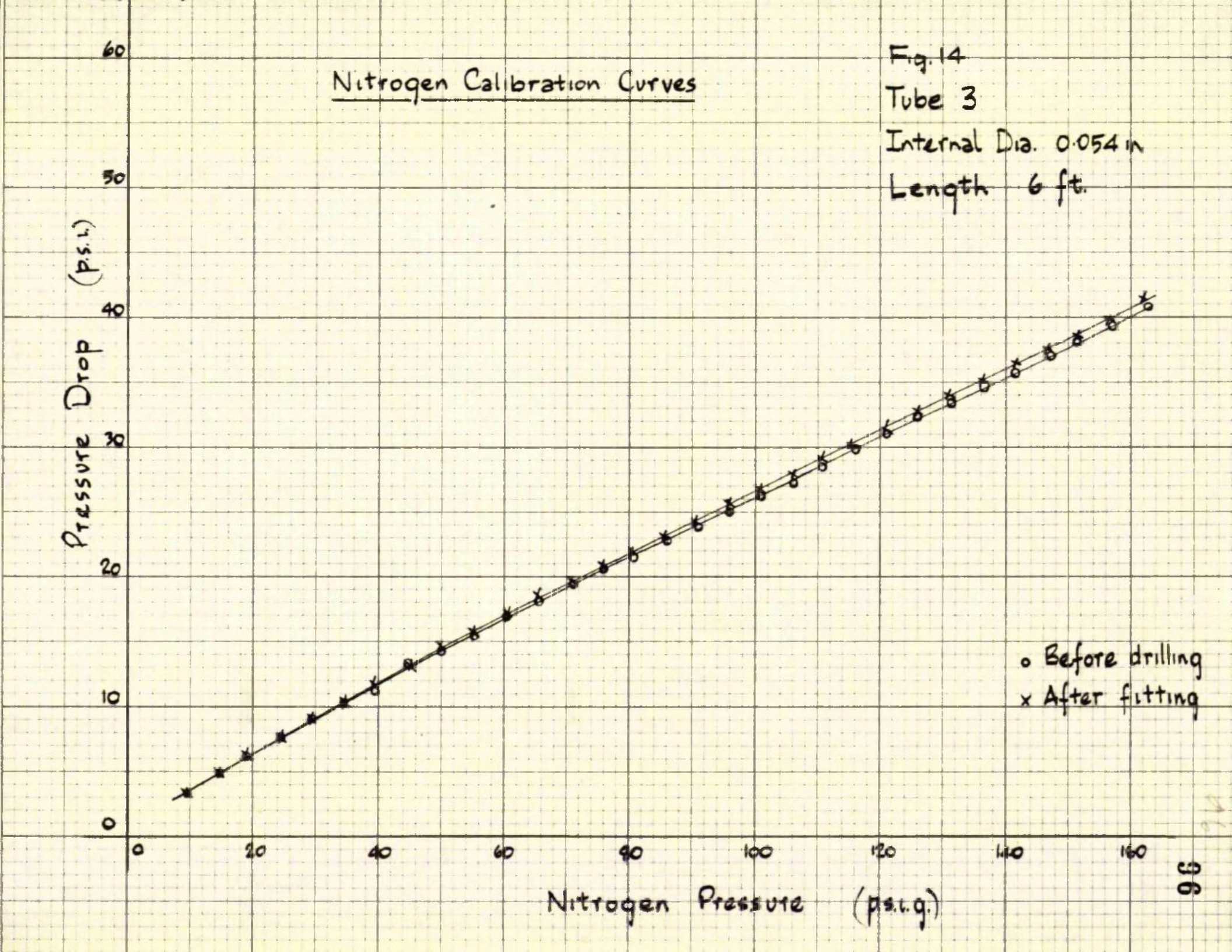
The pressure tap holes were then drilled with a 0.013 in diameter drill, the burrs cleared with a steel piano wire, and the pressure tap connections soldered. The tubes were re-tested for nitrogen pressure drop.

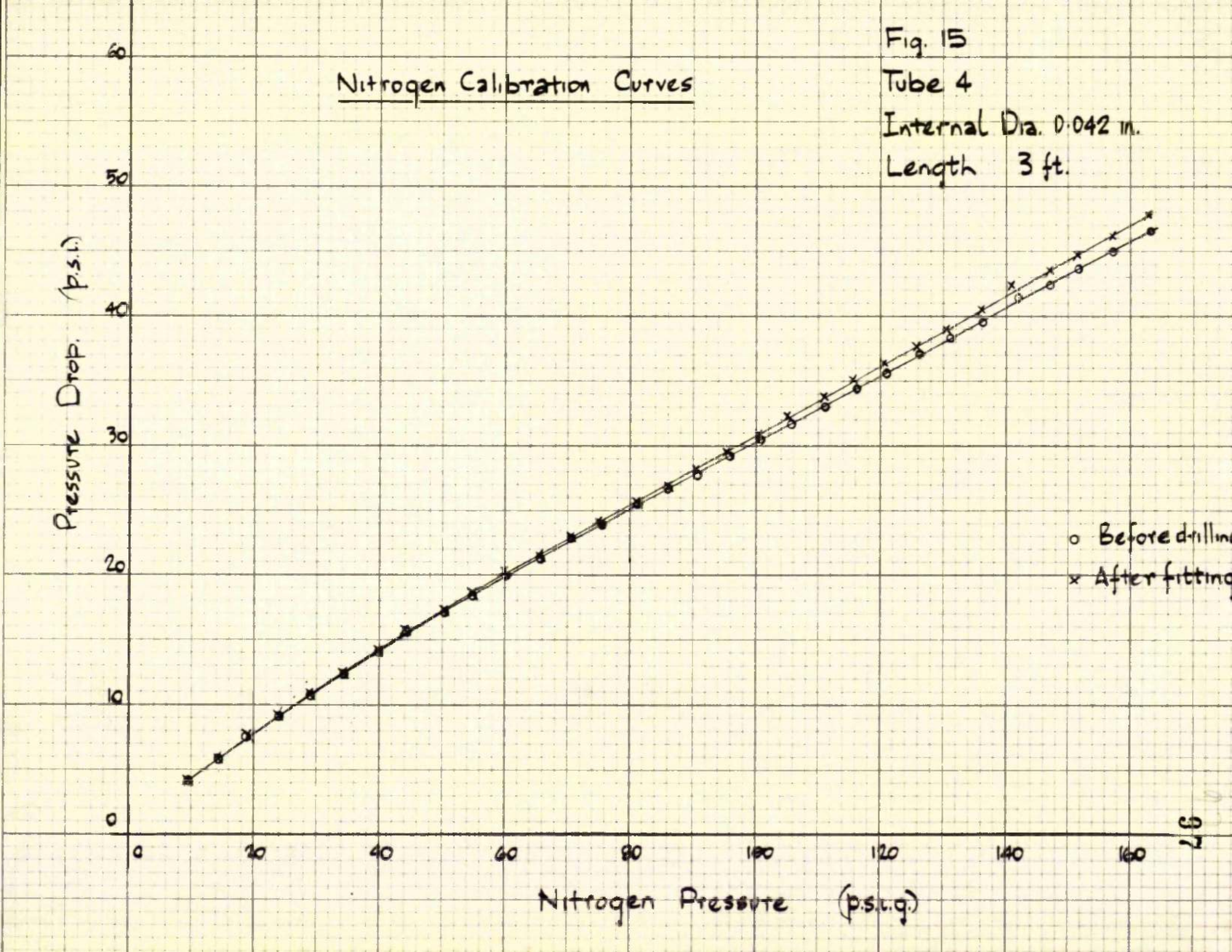
The two pressure drops are plotted on the same graph to enable a comparison, and estimate the effect of boring the holes, on the flow of nitrogen.

It may be seen from these graphs, that the effect was negligible, indicating that the burrs were successfully cleared, leaving the fluid flow virtually unaffected by the pressure tap holes.


Nitrogen Calibration Curves
Pressure Drop (p.s.i.)
0
10
20
30
40
50
60
0
20
40
60
80
100
120
140
160
Nitrogen Pressure (p.s.i.g.)
Fig. 12
Tube 1
Internal Dia. 0.036 in
Length 3 ft.
o Before drilling
× After fitting


Nitrogen Calibration Curves
Fig.13
Tube 2
Inside Dia. 0·054 in.
Length 3 ft.
Pressure Drop (p.s.i.)
0
5
10
15
20
25
0
20
40
60
80
100
120
140
160
Nitrogen Pressure (p.s.i.g.)
o Before drilling
× After fitting


Nitrogen Calibration Curves
Fig. 14
Tube 3
Internal Dia. 0.054 in
Length 6 ft.
Pressure Drop (p.s.i.)
0
10
20
30
40
50
60
0
20
40
60
80
100
120
140
160
Nitrogen Pressure (p.s.i.g.)
o Before drilling
x After fitting


Nitrogen Calibration Curves
Fig. 15
Tube 4
Internal Dia. 0·042 in.
Length 3 ft.
Pressure Drop (p.s.i.)
Nitrogen Pressure (p.s.i.g.)
0
10
20
30
40
50
60
0
20
40
60
80
100
120
140
160
o Before drilling
x After fitting

Nitrogen Calibration Curves

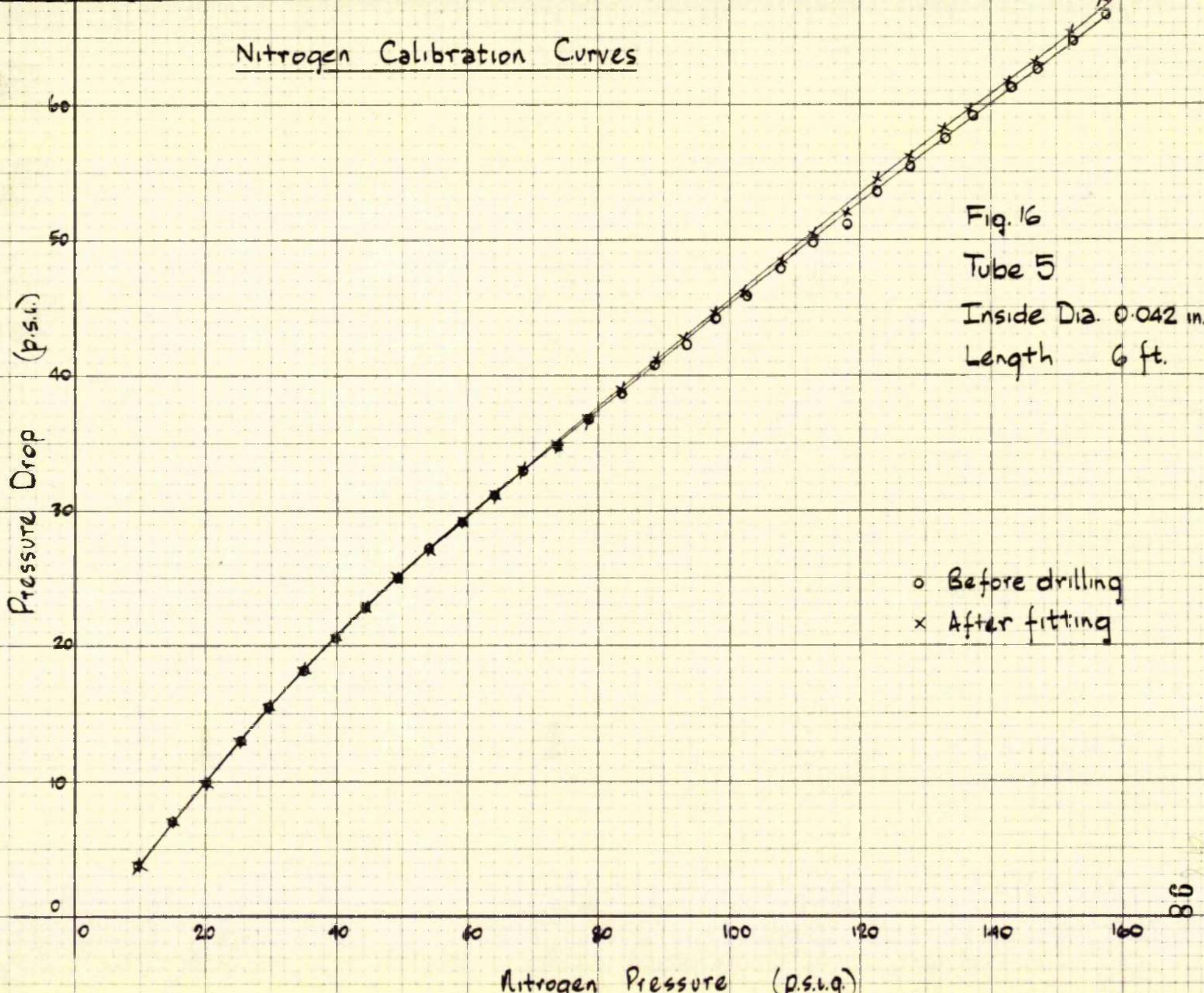


Fig. 16

Tube 5

Inside Dia. 0·042 in.

Length 6 ft.

o Before drilling

× After fitting

9.2 Pressure and Temperature Distribution Curves

Pressure and temperature were observed along the tube length, at locations given in section 7.6, and which are also marked in the following graphs.

While the pressure drops in a pattern normally expected, the "temperature drop" curves show an increase in temperature as the solution flows along the tube, decreasing in the later sections of the tube. The possible causes for this are discussed in a later chapter.

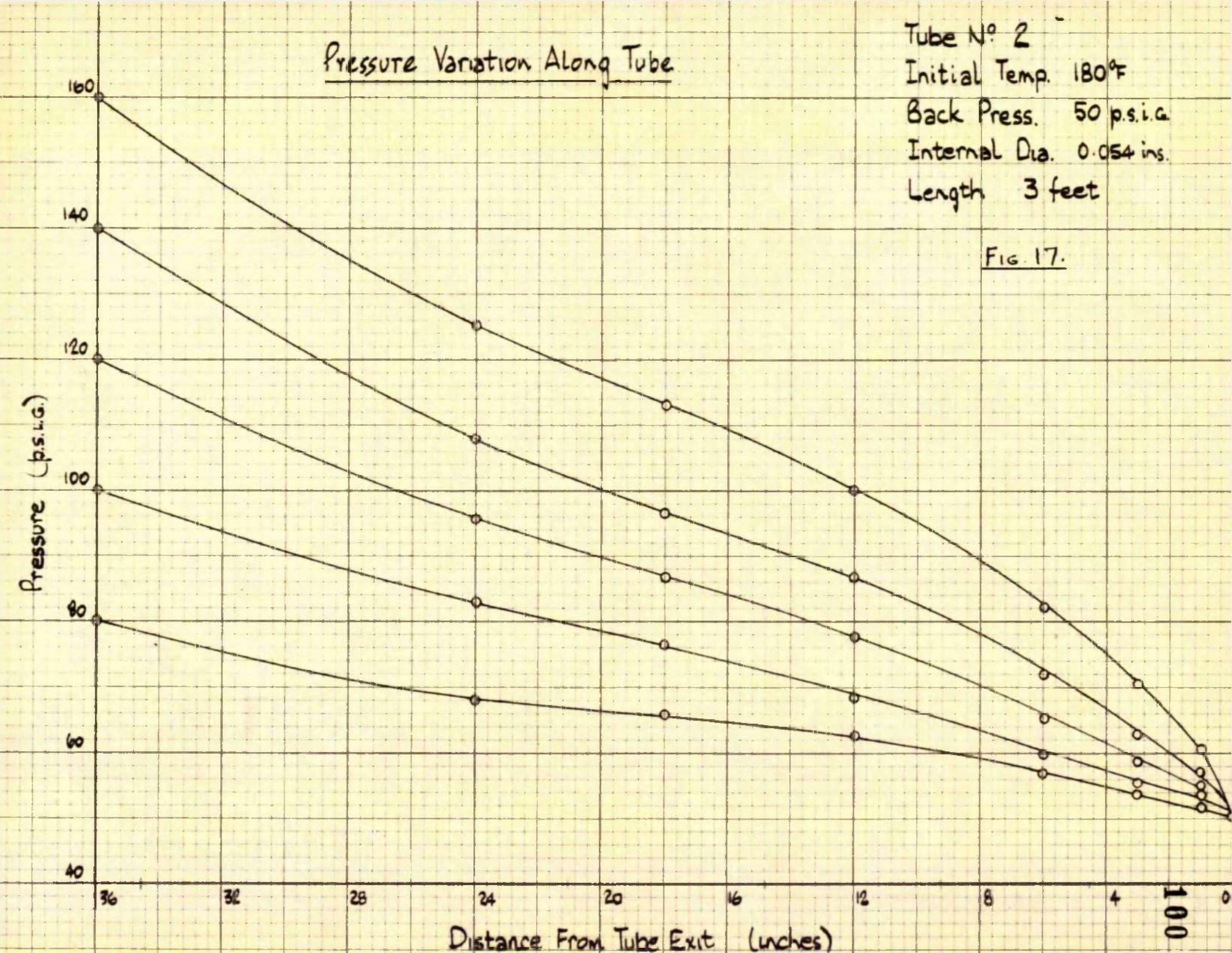
It may be noticed from these graphs that the temperature remains above the saturation temperature for R12, throughout the entire flow, and therefore any free R12 is confined to the vapour phase.

Pressure and temperature measurements were made on all tubes tested. Since the pattern was similar, graphs are presented here for two tubes only.

Pressure Variation Along Tube

Tube No. 2
Initial Temp. 180°F
Back Press. 50 p.s.i.g.
Internal Dia. 0.054 ins.
Length 3 feet

Fig. 17.



Temperature & Pressure Variation

Along Tube

Tube N° 2
Initial Temp. 180°F.
Back Press. 30 p.s.i.g.
Internal Dia. 0.054 in.
Length 3 ft.

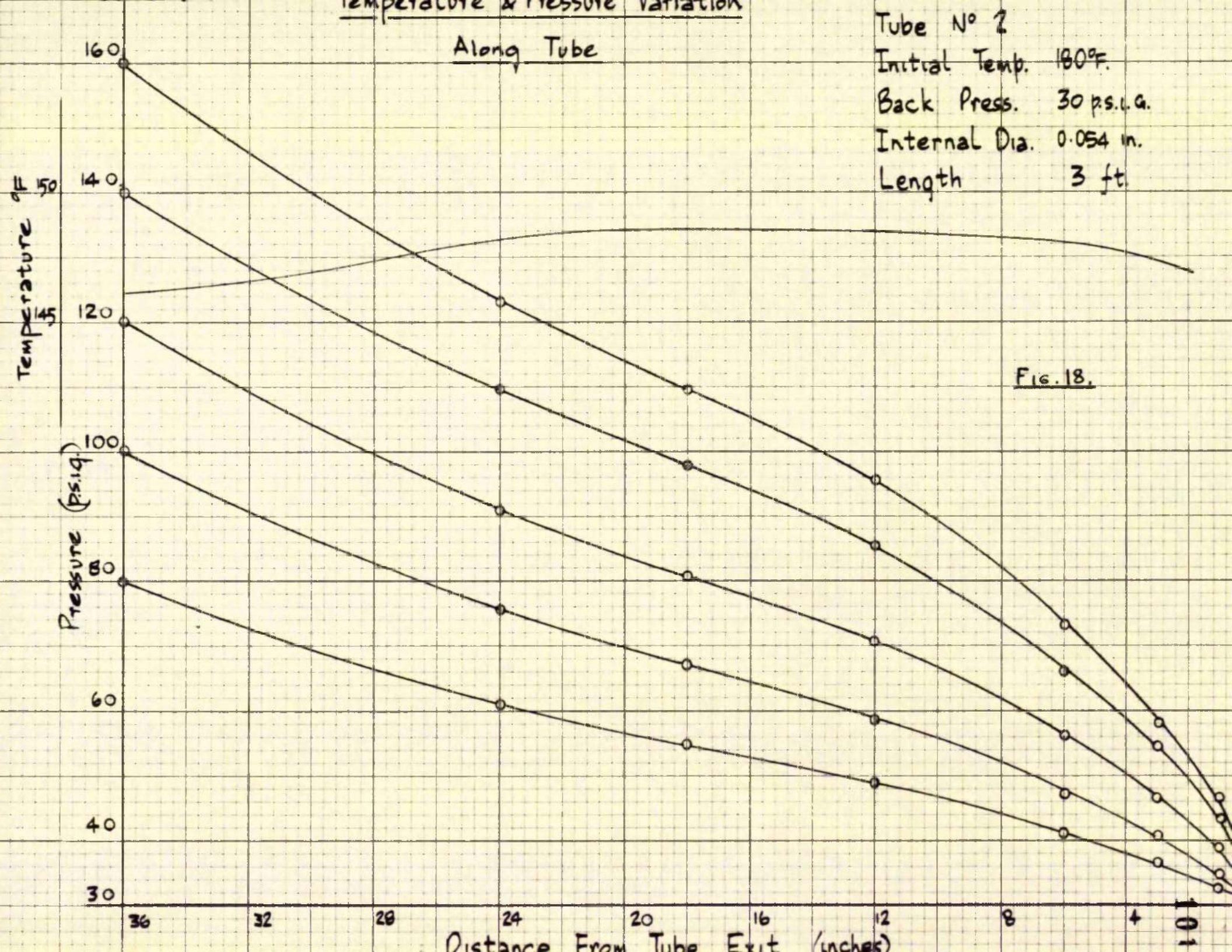
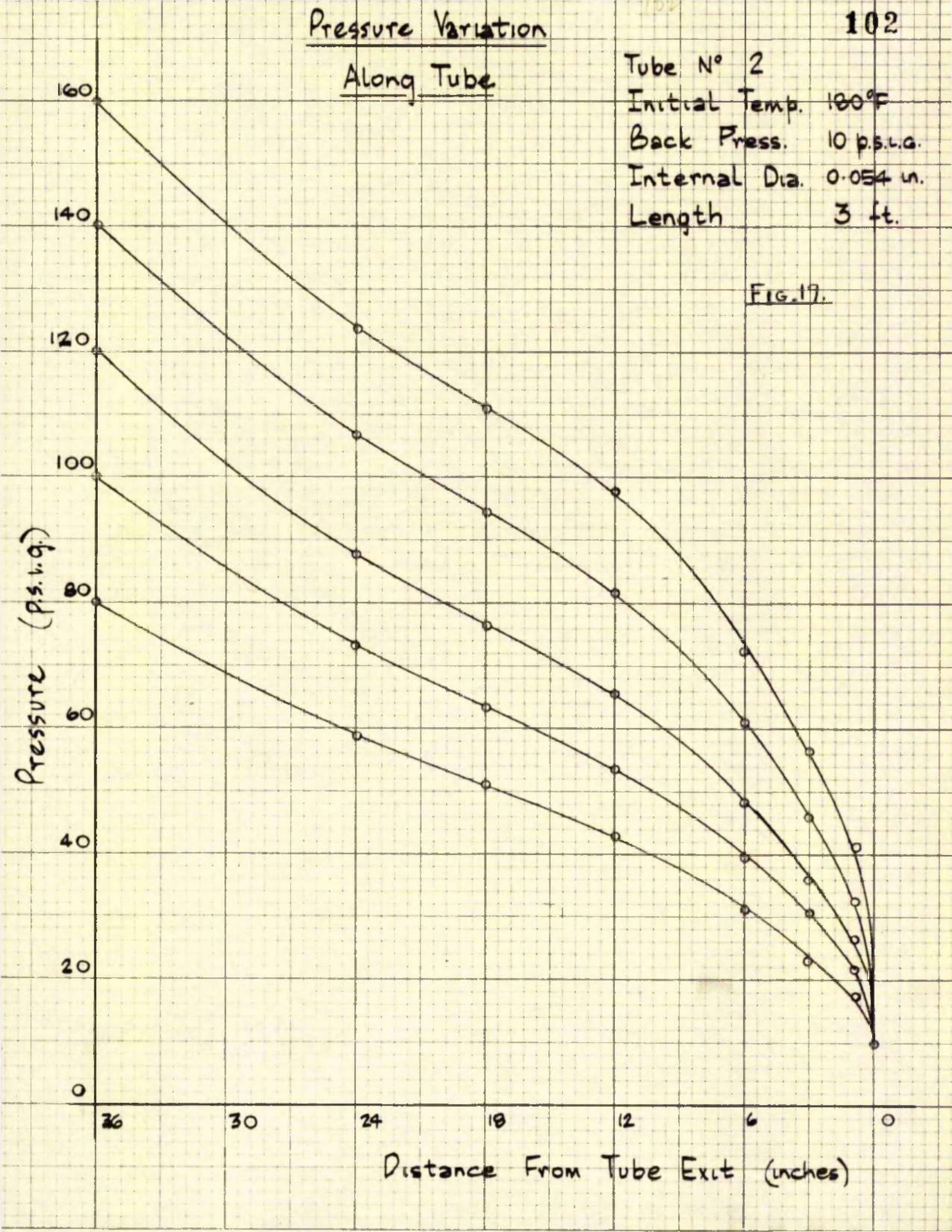


Fig. 18.

Pressure Variation Along Tube

Tube N° 2
Initial Temp. 180°F
Back Press. 10 p.s.i.g.
Internal Dia. 0.054 in.
Length 3 ft.

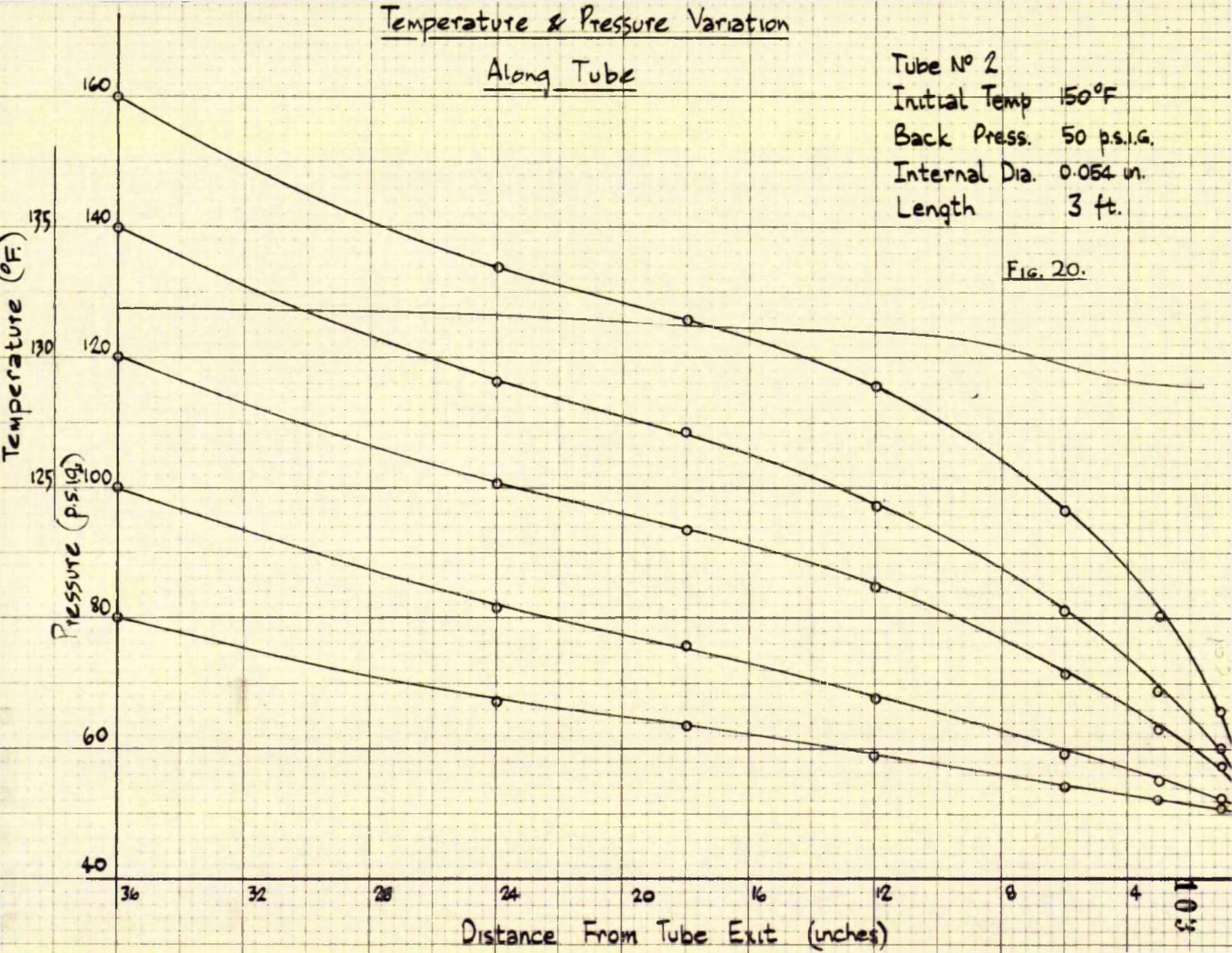
Fig. 17.

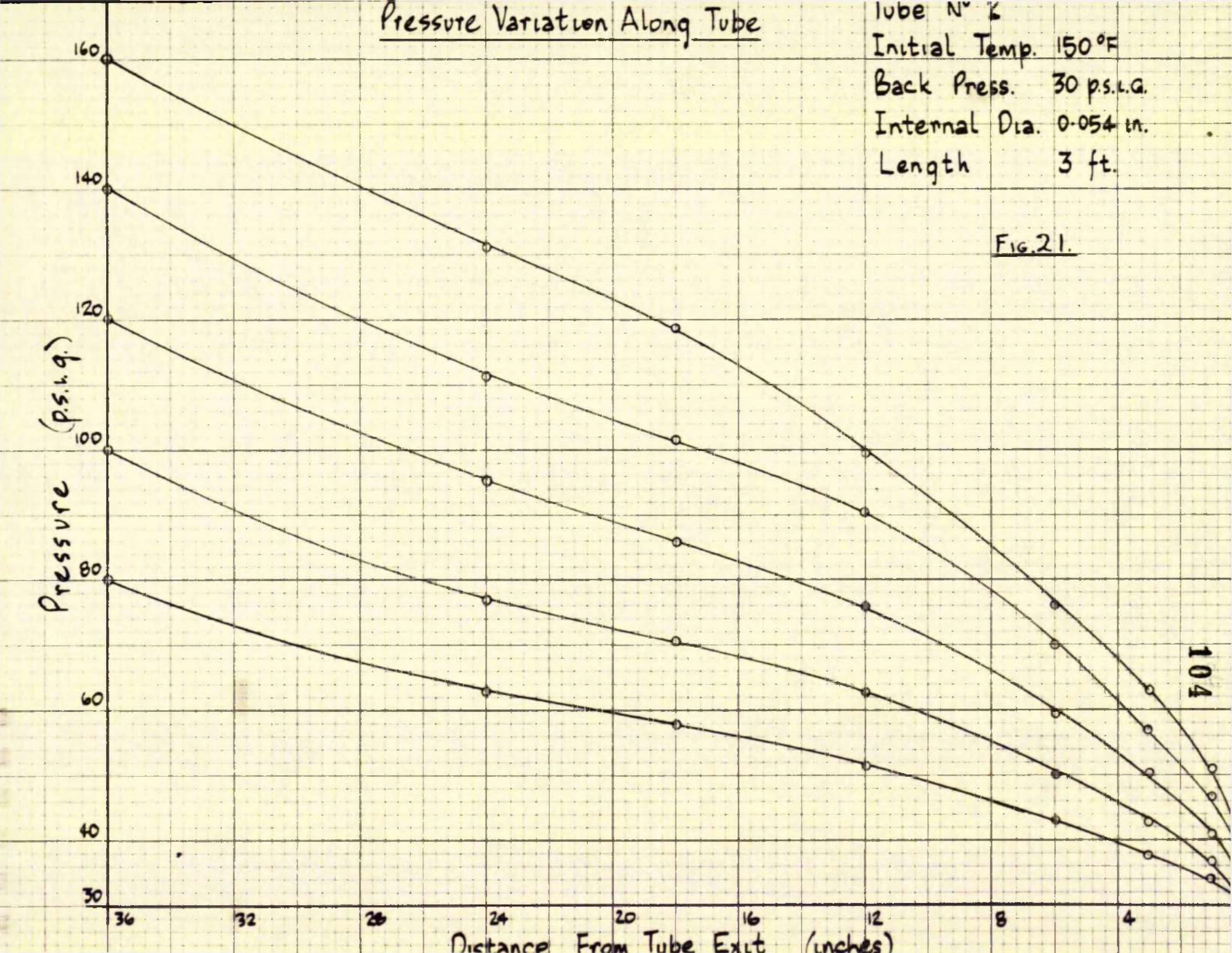


Temperature & Pressure Variation Along Tube

Tube N° 2
Initial Temp 150°F
Back Press. 50 p.s.i.g.
Internal Dia. 0.054 in.
Length 3 ft.

Fig. 20.

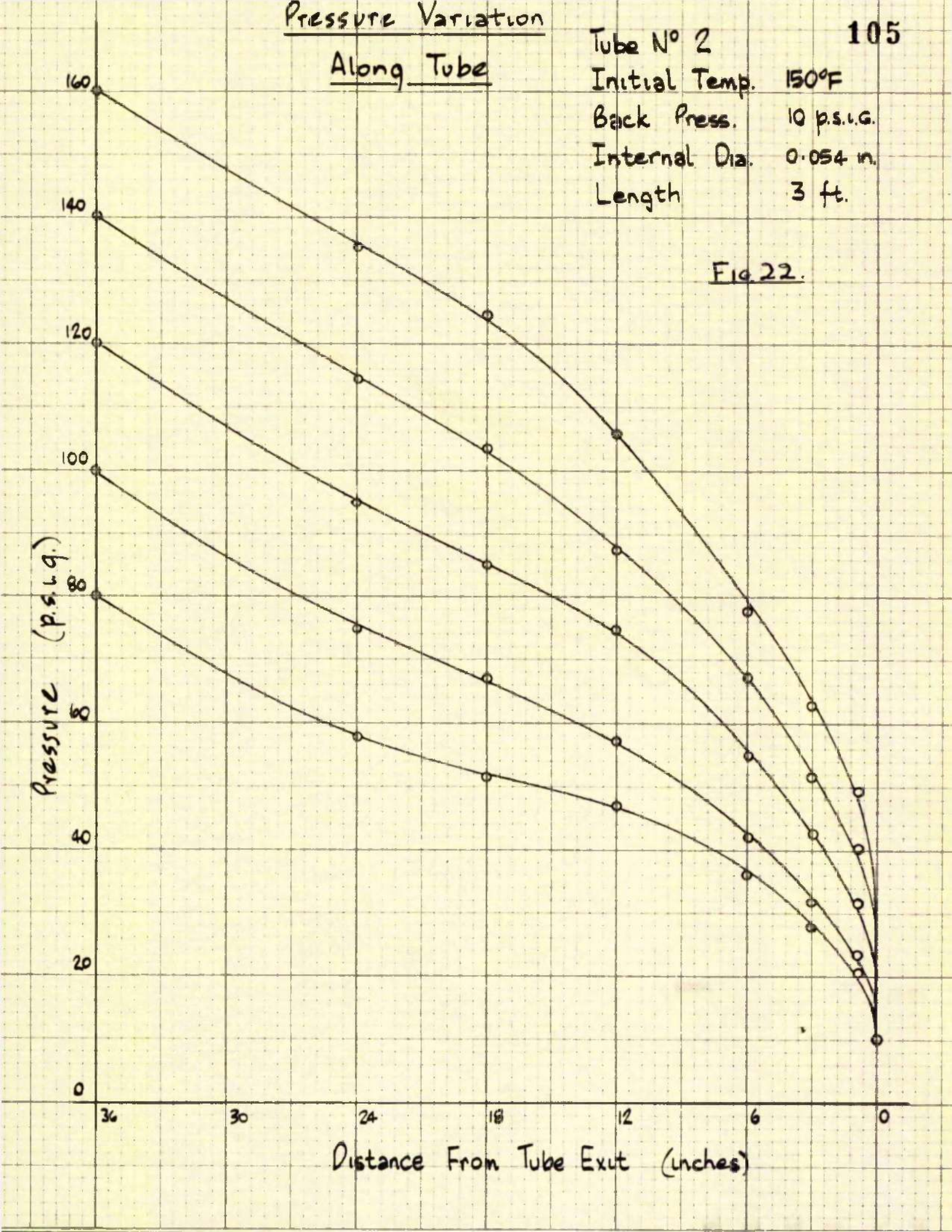



Pressure Variation Along Tube
Tube N° 2
Initial Temp. 150°F
Back Press. 30 p.s.i.g.
Internal Dia. 0·054 in.
Length 3 ft.
Fig. 21.
Pressure (p.s.i.g.)
160
140
120
100
80
60
40
30
36
32
28
24
20
16
12
8
4
Distance From Tube Exit (inches)
104

Pressure Variation Along Tube

Tube N° 2
Initial Temp. 150°F
Back Press. 10 p.s.i.g.
Internal Dia. 0·054 in.
Length 3 ft.

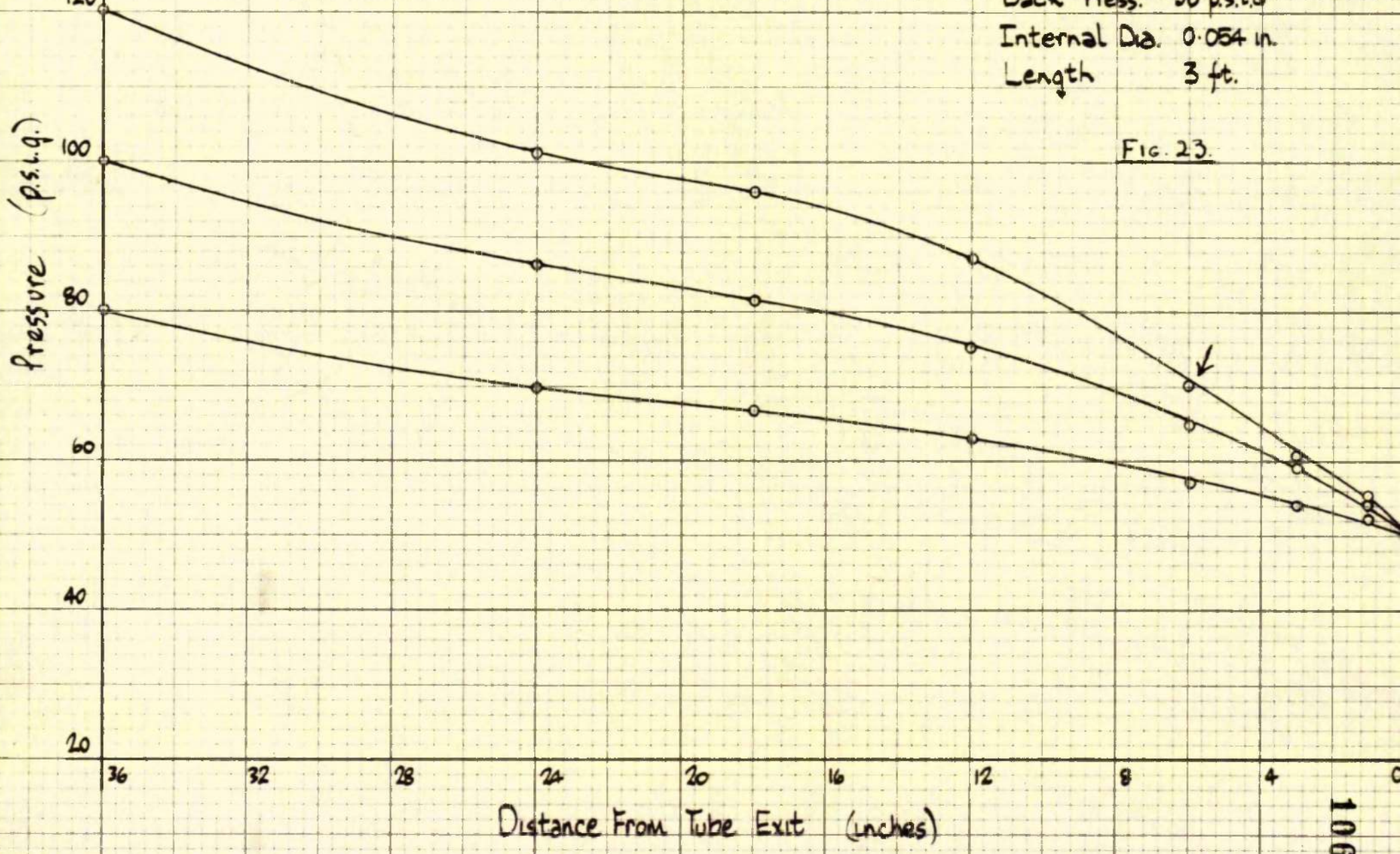
Fig 22.

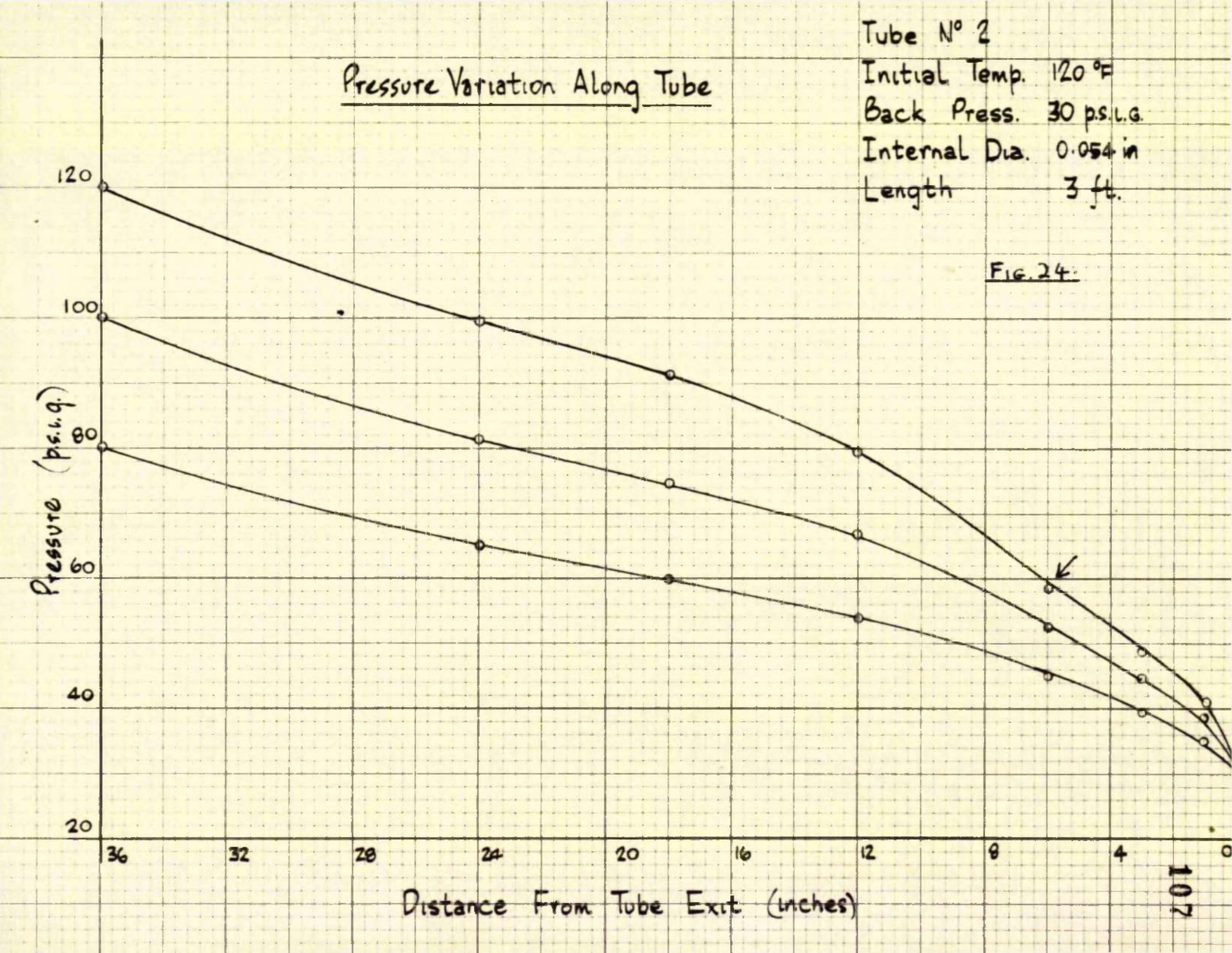


Pressure Variation Along Tube

Tube N° 2
Initial Temp. 120°F
Back Press. 50 p.s.i.g
Internal Dia. 0·054 in.
Length 3 ft.

Fig. 23.

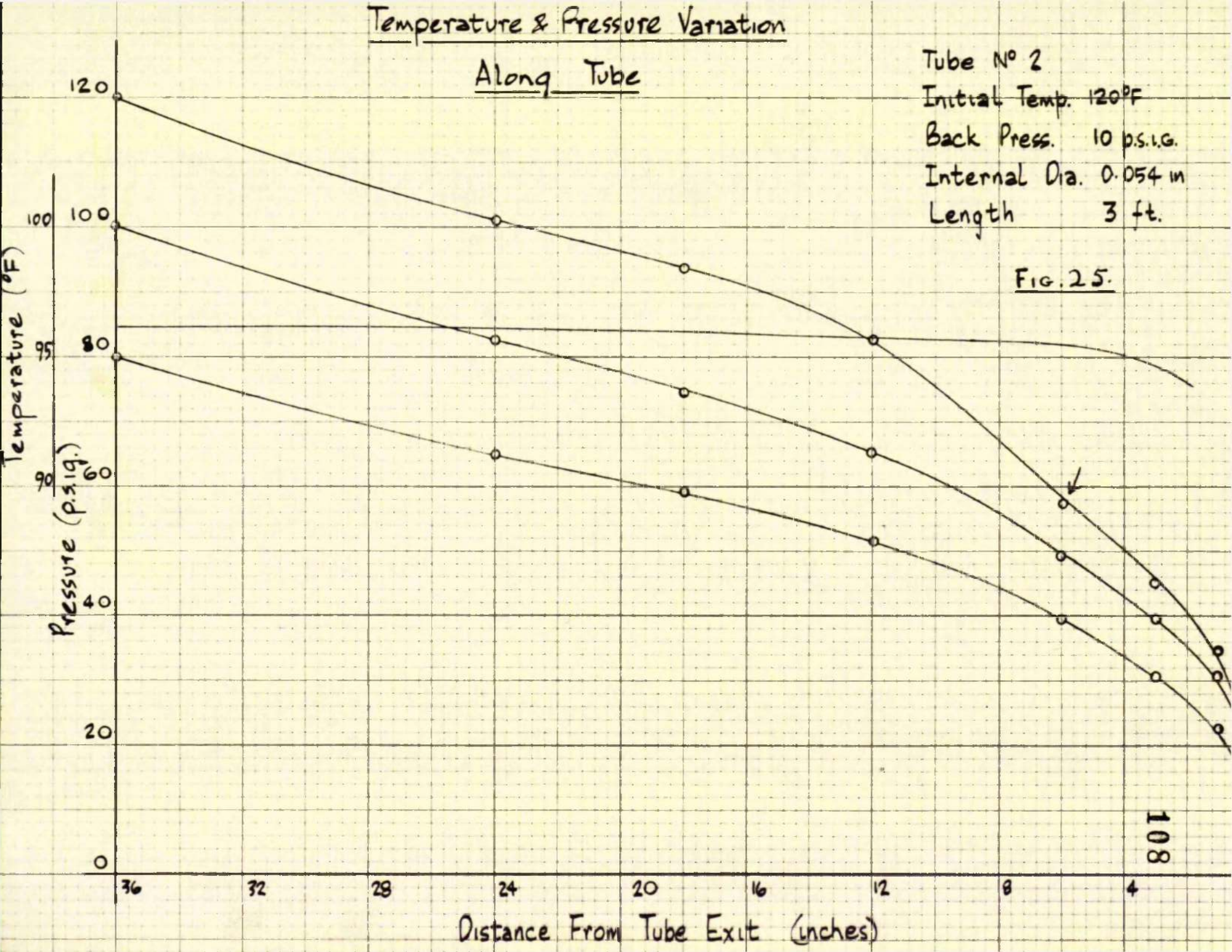



Pressure Variation Along Tube
Tube N° 2
Initial Temp. 120 °F
Back Press. 30 p.s.i.g.
Internal Dia. 0.054 in
Length 3 ft.
Fig. 24.
Pressure (p.s.i.g.)
120
100
80
60
40
20
36
32
28
24
20
16
12
8
4
0
Distance From Tube Exit (inches)
201

Temperature & Pressure Variation Along Tube

Tube N° 2
Initial Temp. 120°F
Back Press. 10 p.s.i.g.
Internal Dia. 0.054 in
Length 3 ft.

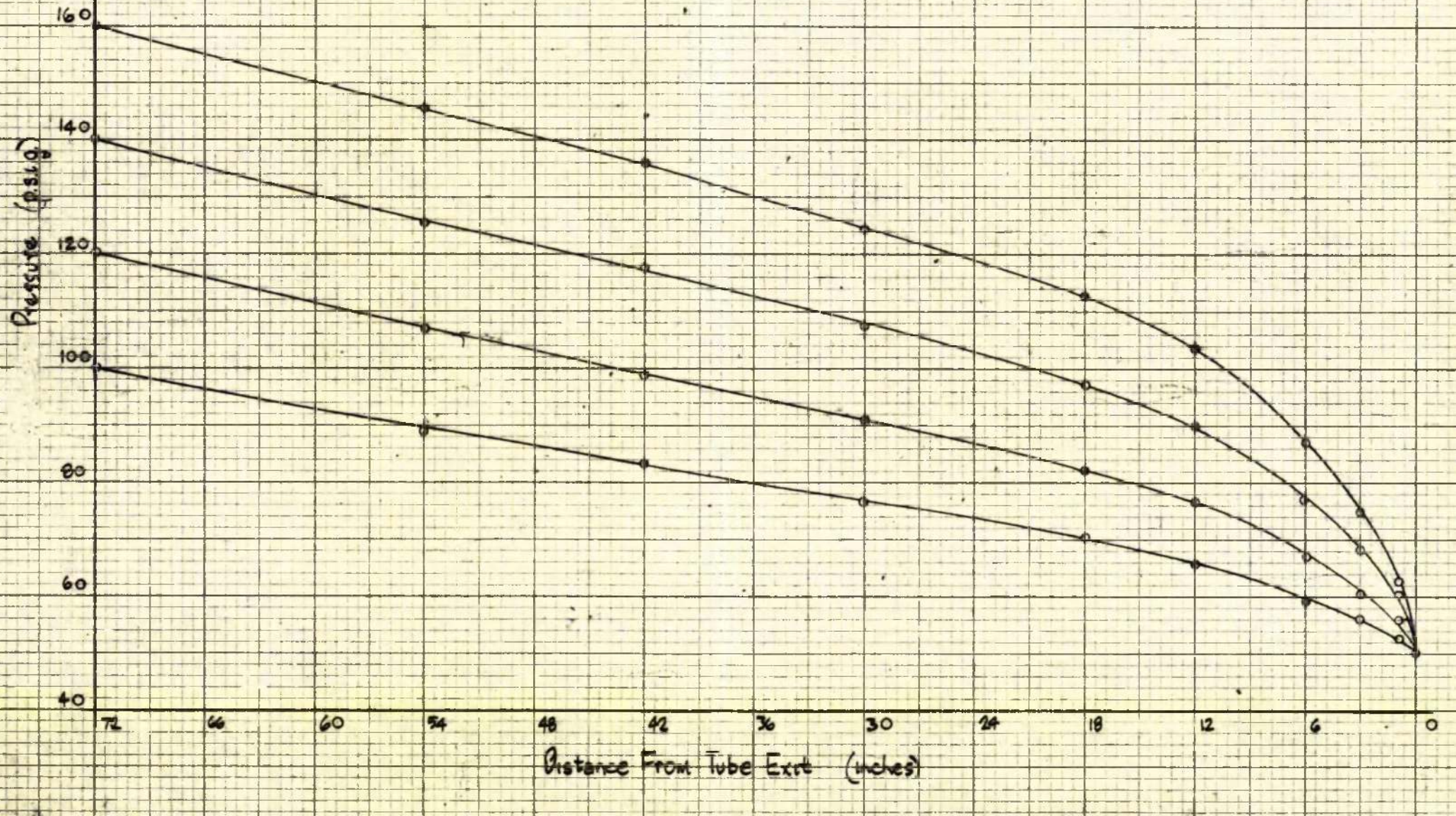
Fig. 25.



Pressure Variation Along Tube

Tube No. 3
Initial Temp. 190°F
Back Press. 50 p.s.i.g.
Internal Dia. 0.054 in.
Length 6 ft.

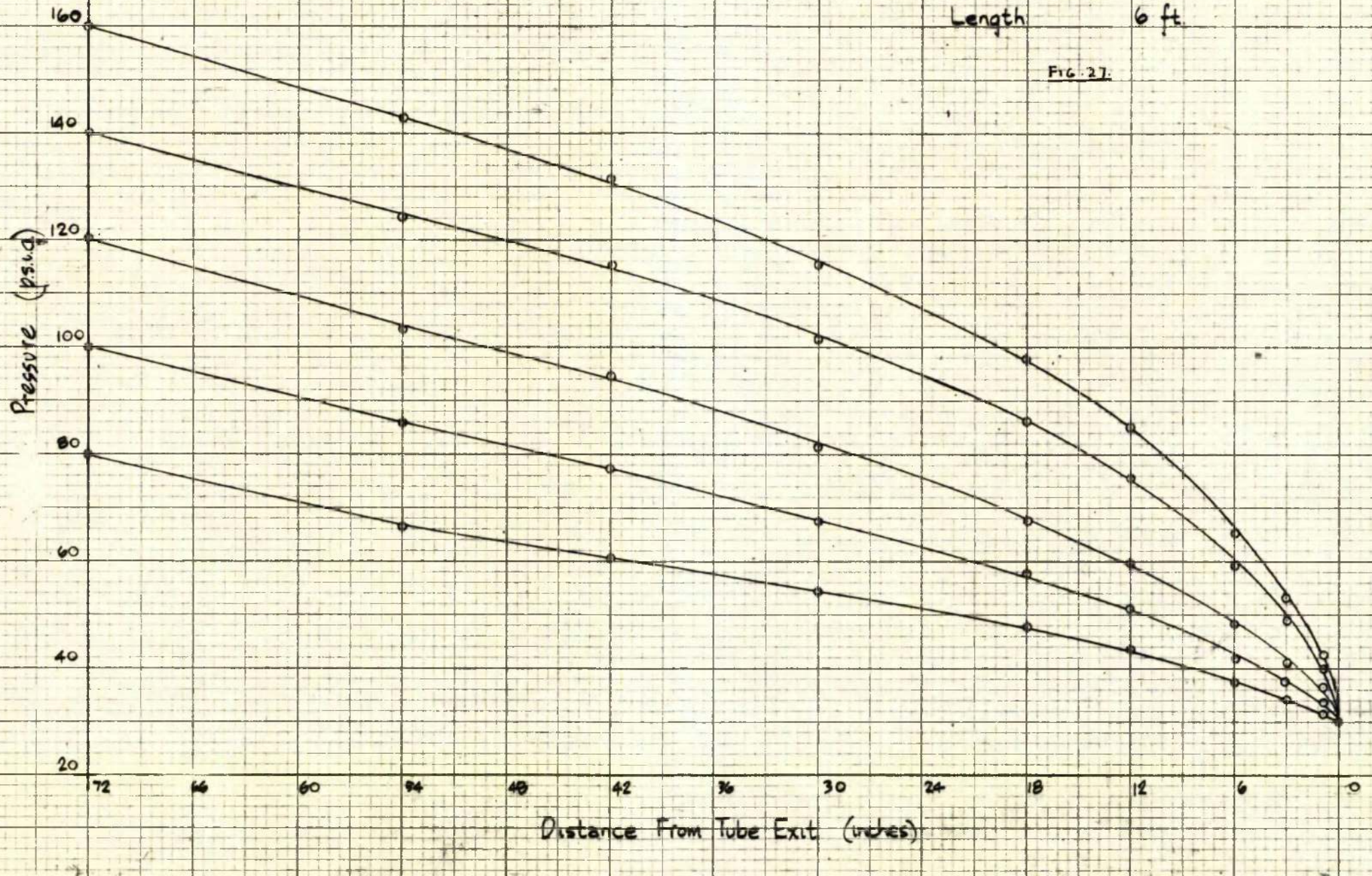
Fig. 26.



Pressure Variation Along Tube

Tube N° 3
Initial Temp. 180°F
Back Press. 30 psi.g.
Internal Dia. 0.054 in.
Length 6 ft.

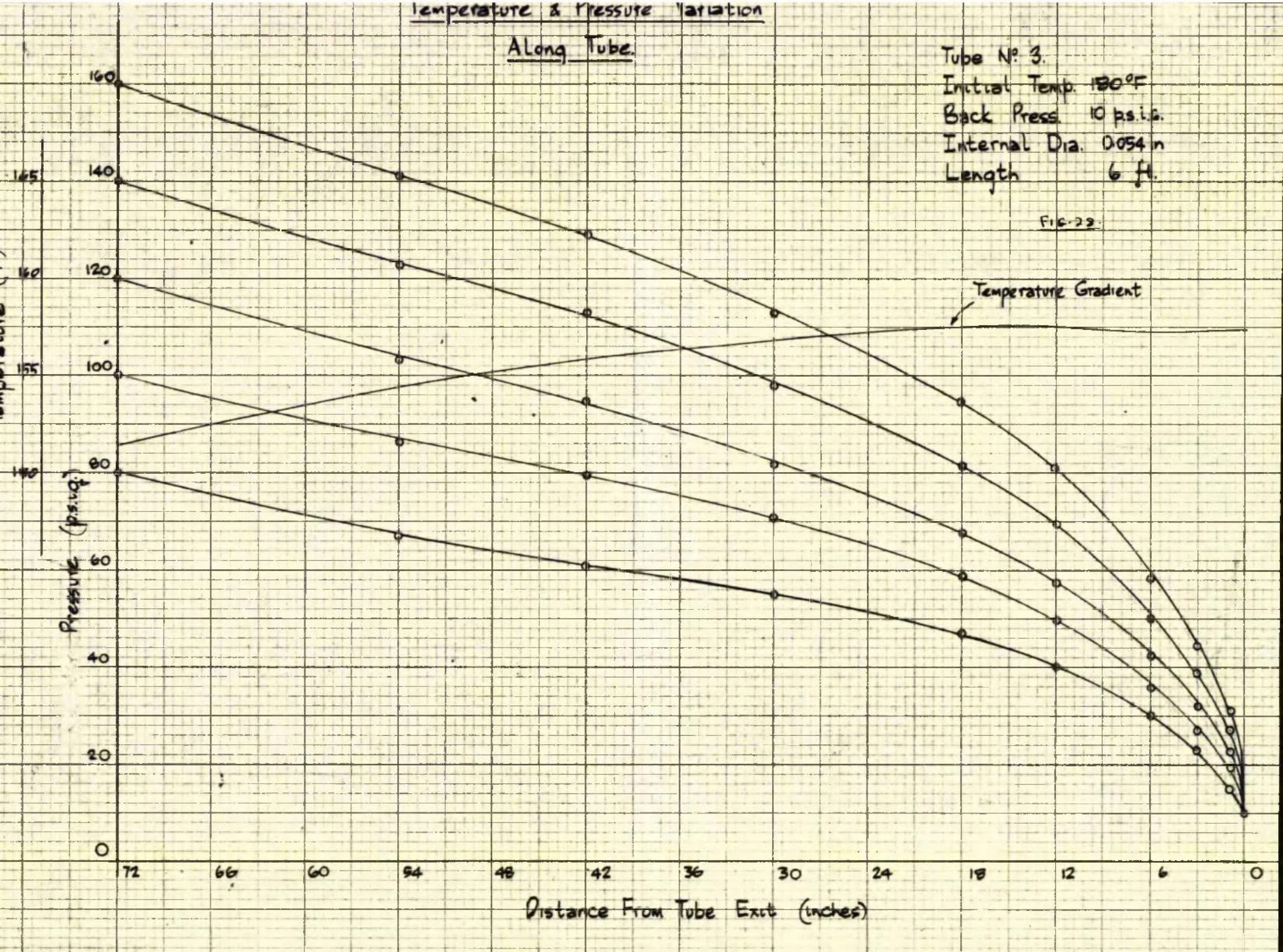
FIG. 27.

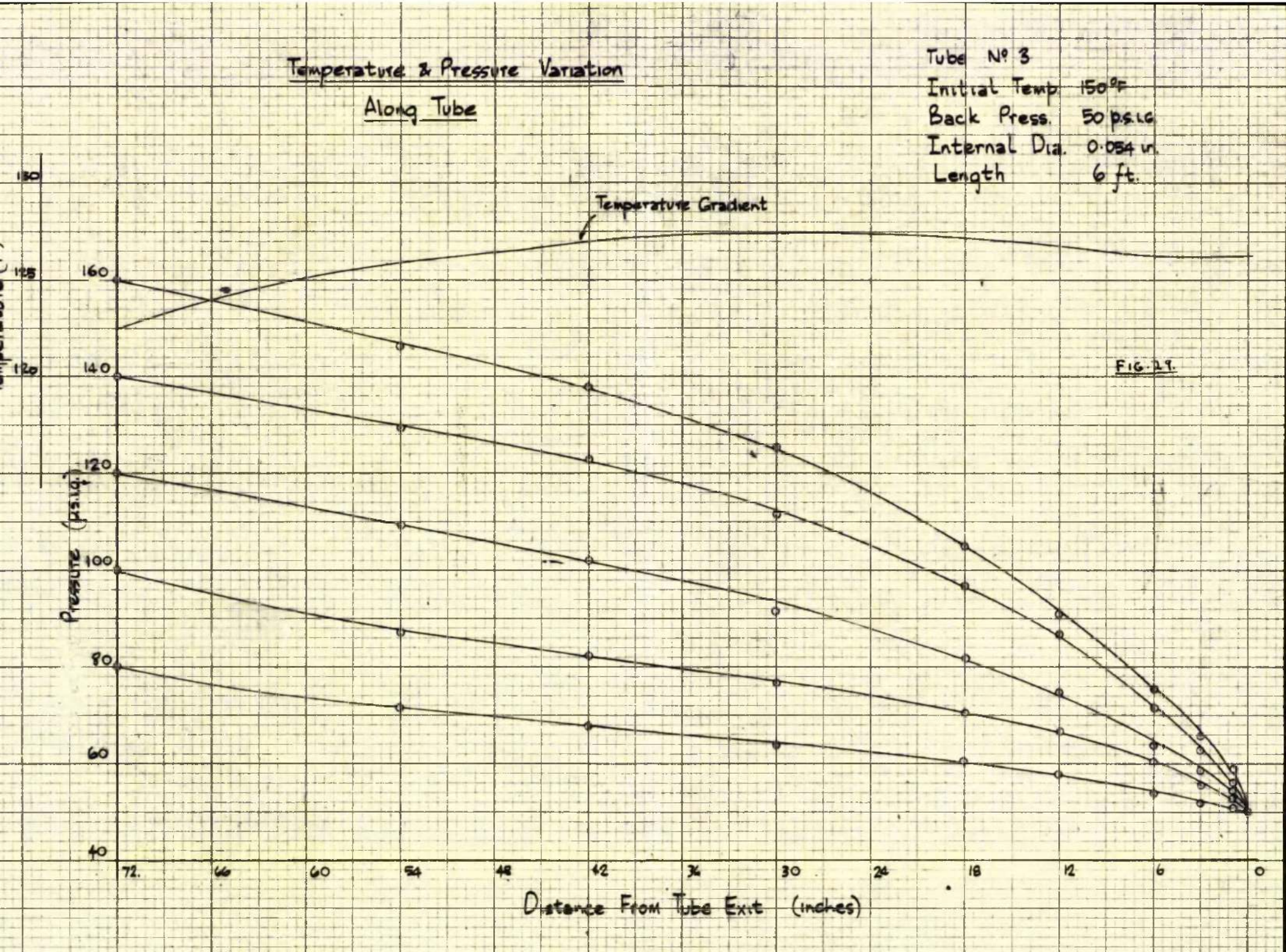


Temperature & Pressure Variation Along Tube.

Tube No. 3.
Initial Temp. 180°F
Back Press. 10 p.s.i.g.
Internal Dia. .0054 in
Length 6 ft.

Fig. 22.




Temperature & Pressure Variation
Along Tube
Tube Nº 3
Initial Temp 150°F
Back Press. 50 p.s.i.g
Internal Dia. 0.054 in.
Length 6 ft.
Temperature Gradient
FIG. 29.
Temperature (°F)
Pressure (p.s.i.g.)
Distance From Tube Exit (inches)
72.
66
60
54
48
42
36
30
24
18
12
6
0
160
140
120
100
80
60
40
150
125
120

Pressure Variation Along Tube

Tube N° 3
Initial Temp. 150°F
Back Press. 30 p.s.i.g.
Internal Dia 0.054 in.
Length 6 ft.

Fig 30.

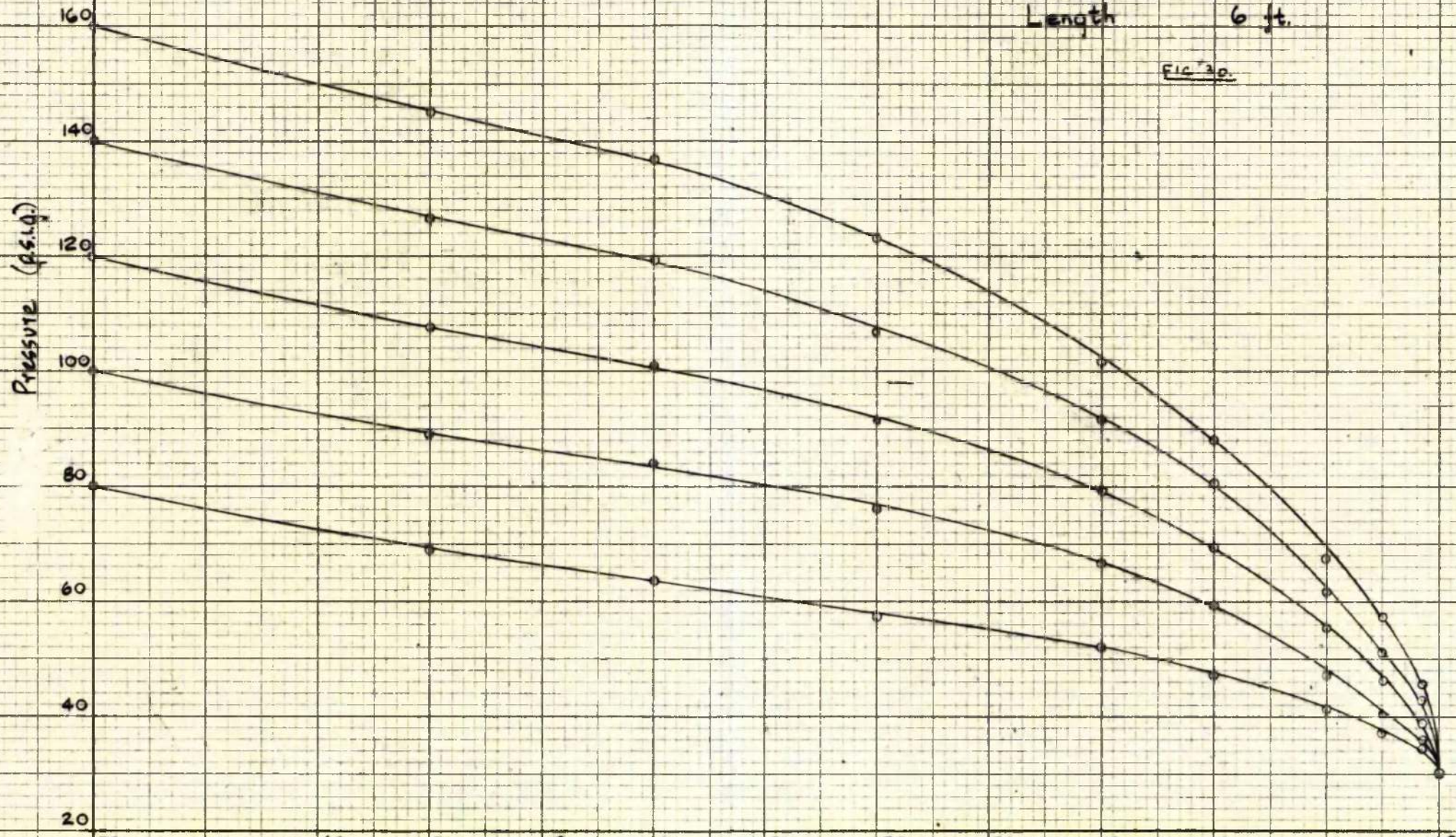


Fig. 30.

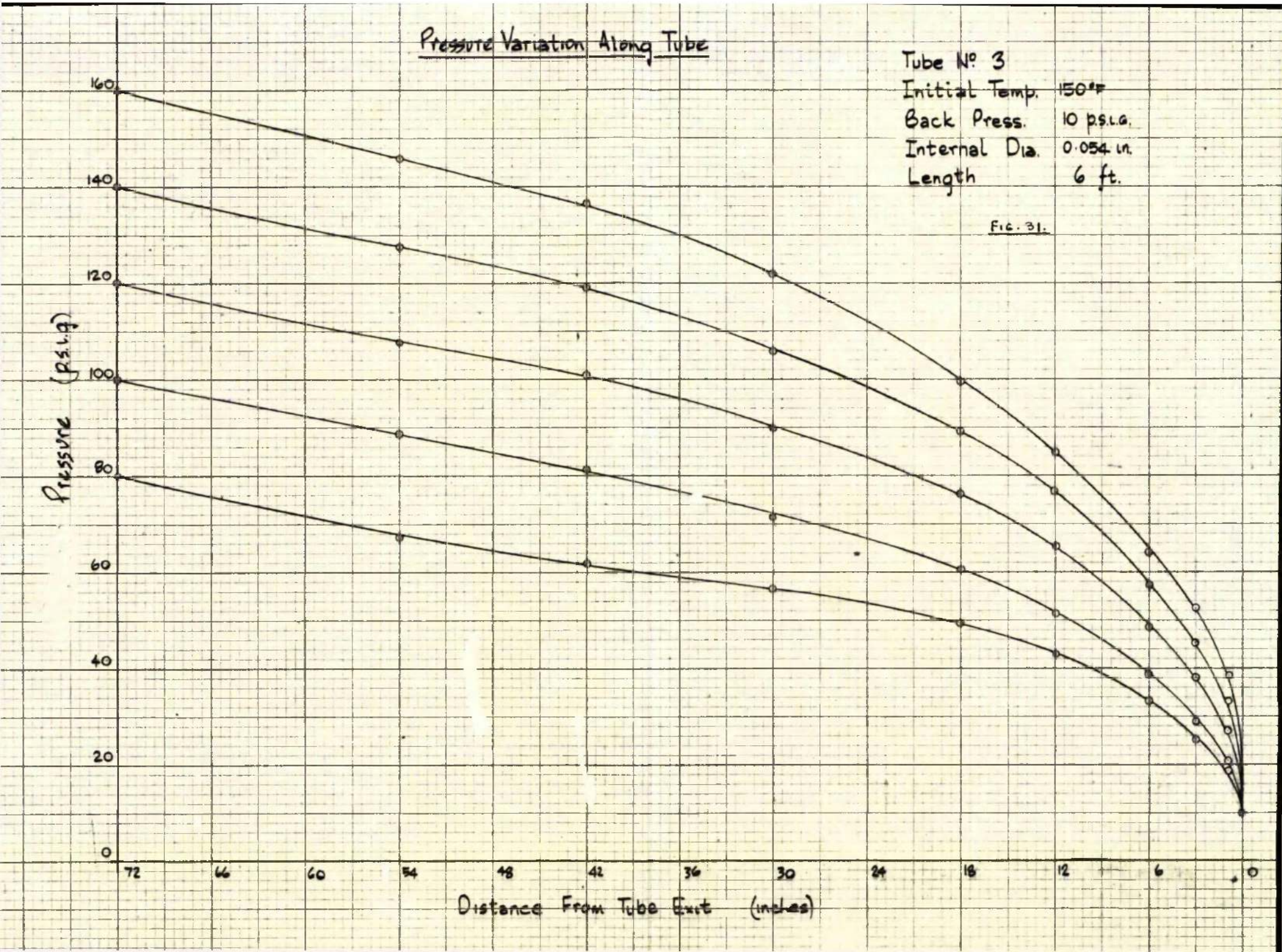
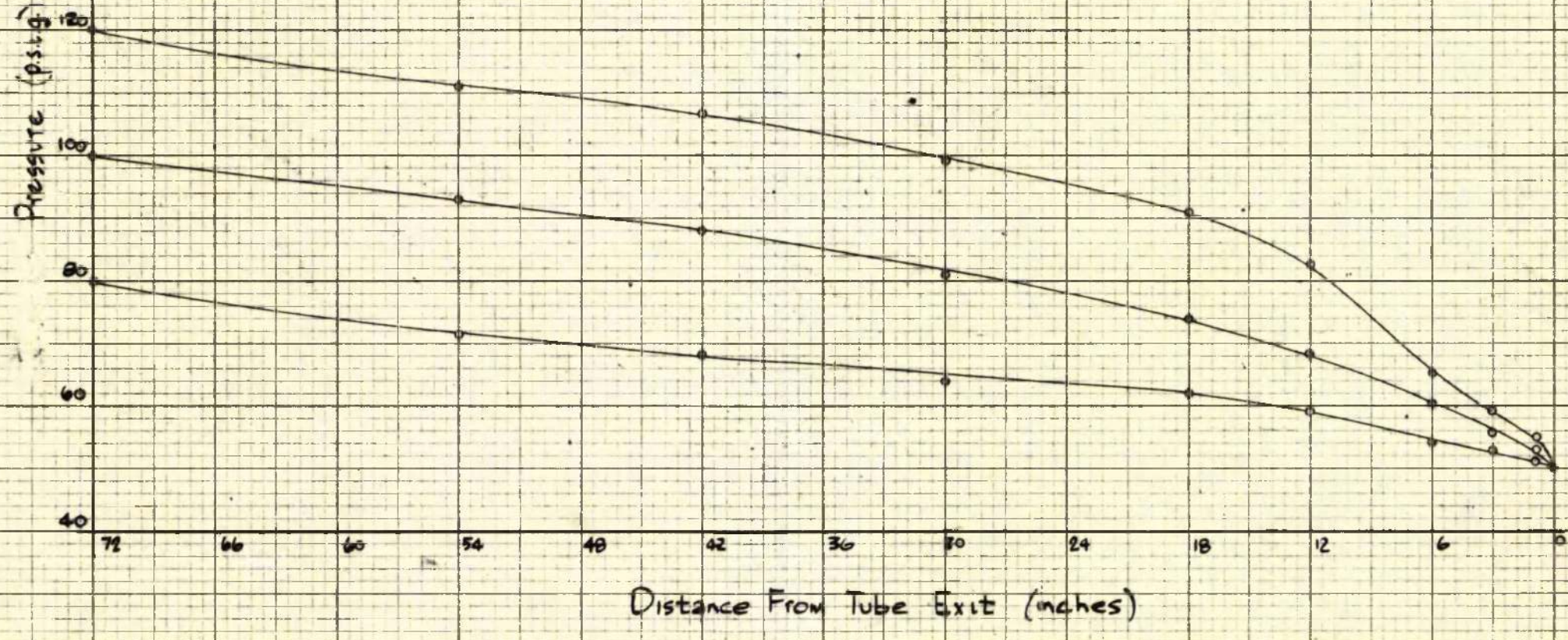

Pressure Variation Along Tube
Tube No. 3
Initial Temp. 150°F
Back Press. 10 p.s.i.g.
Internal Dia. 0.054 in.
Length 6 ft.
Fig. 31.
Pressure (p.s.i.g.)
160
140
120
100
80
60
40
20
0
72
66
60
54
48
42
36
30
24
18
12
6
0
Distance From Tube Exit (inches)

FIG. 31.

Pressure Variation Along Tube

Tube Nº 3
Initial Temp. 120°F
Back Press. 50 p.s.i.g.
Internal Dia. 0.054"
Length 6 Ft.

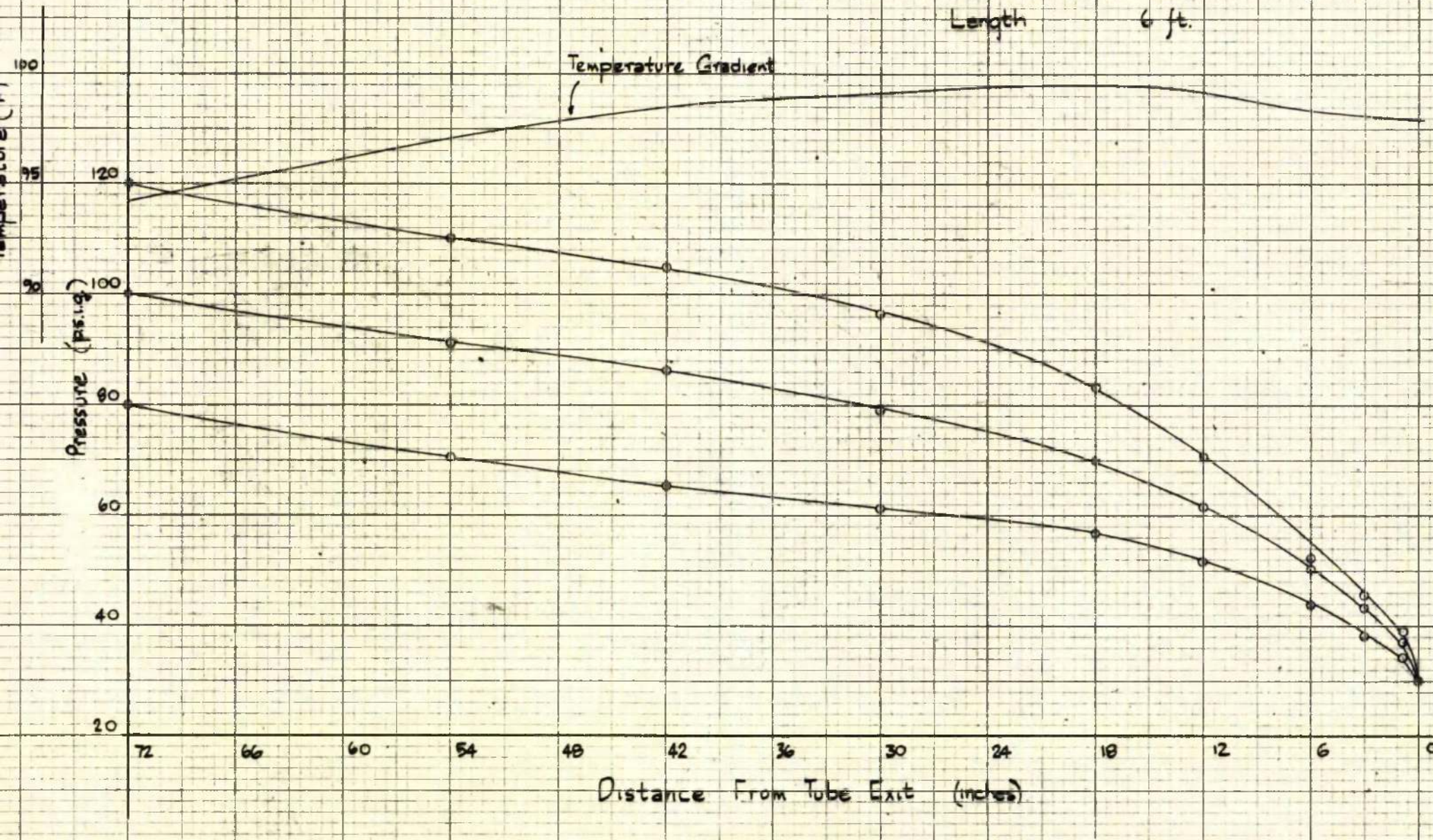
Fig. 32.



Temperature & Pressure Variation Along Tube

Tube N° 3
Initial Temp. 120°F
Back Press. 30 psig
Internal Dia. 0.054 in
Length 6 ft.

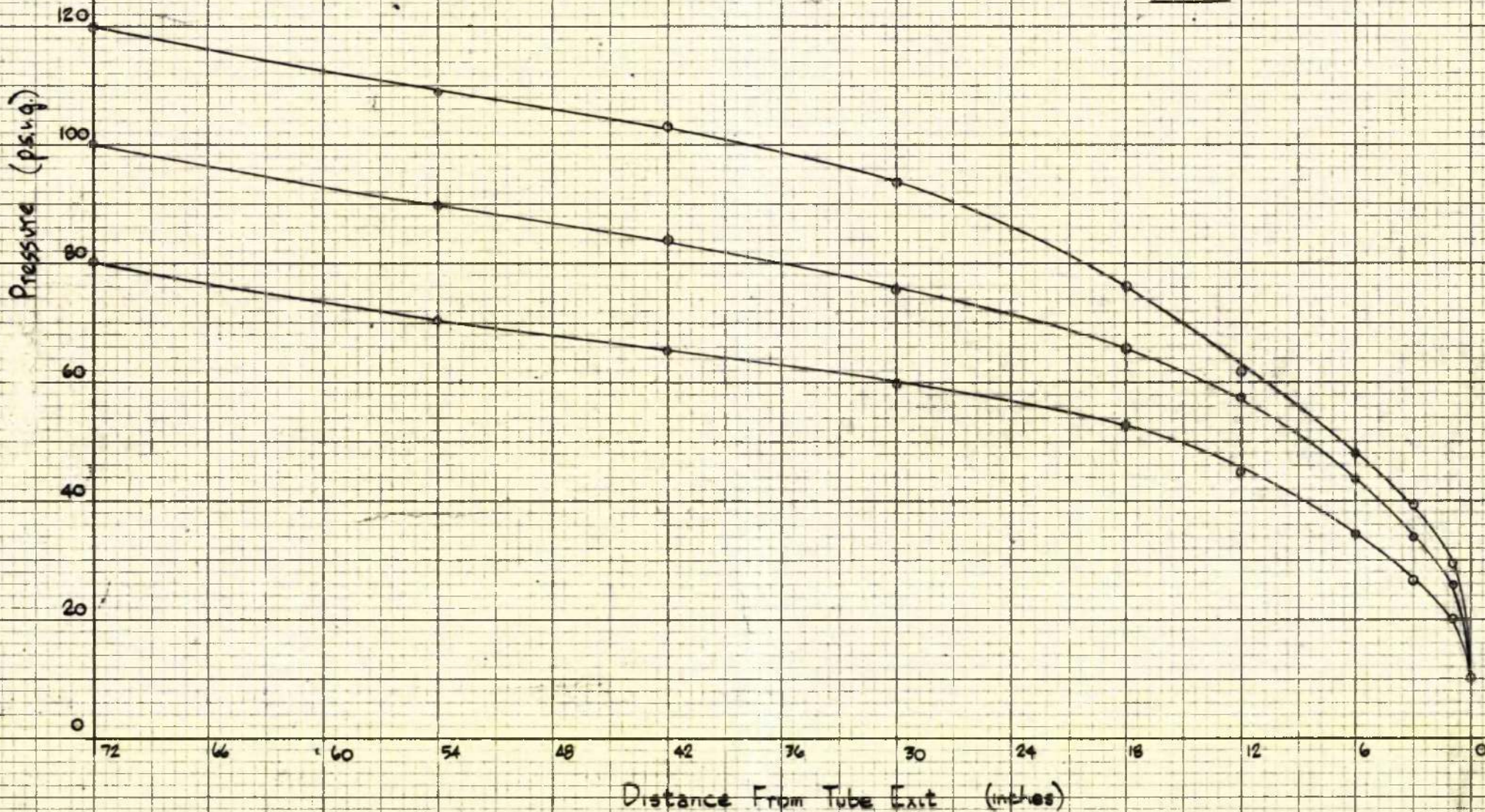
Fig 33.



Pressure Variation Along Tube

Tube N° 3
Initial Temp. 120°F.
Back Press. 10 p.s.i.g.
Internal Dia. 0.054 in.
Length 6 ft.

Fig. 34.



9.3 Variation of Flow Rates with Head and Back Pressure

The individual mass flow rates of oil and of R12 as components of the solution, and the total mass flow rate of solution are presented in graphical form as functions of head and back pressures, and initial temperature.

The total mass of solution is the sum of the oil and of R12 separately collected at the end of each test. The oil collected in the oil receiver is a saturated solution at that pressure and temperature, and the percentage concentration is taken from the solubility curve, Fig. 30. The density of the solution is taken from the density-R12 concentration-temperature relation presented by Löffler (21), Fig. 52. It is not known if the oil used by Löffler was the same as Capella D, but any difference in densities is likely to be small. This was the only published information on the density of oil-R12 solutions.

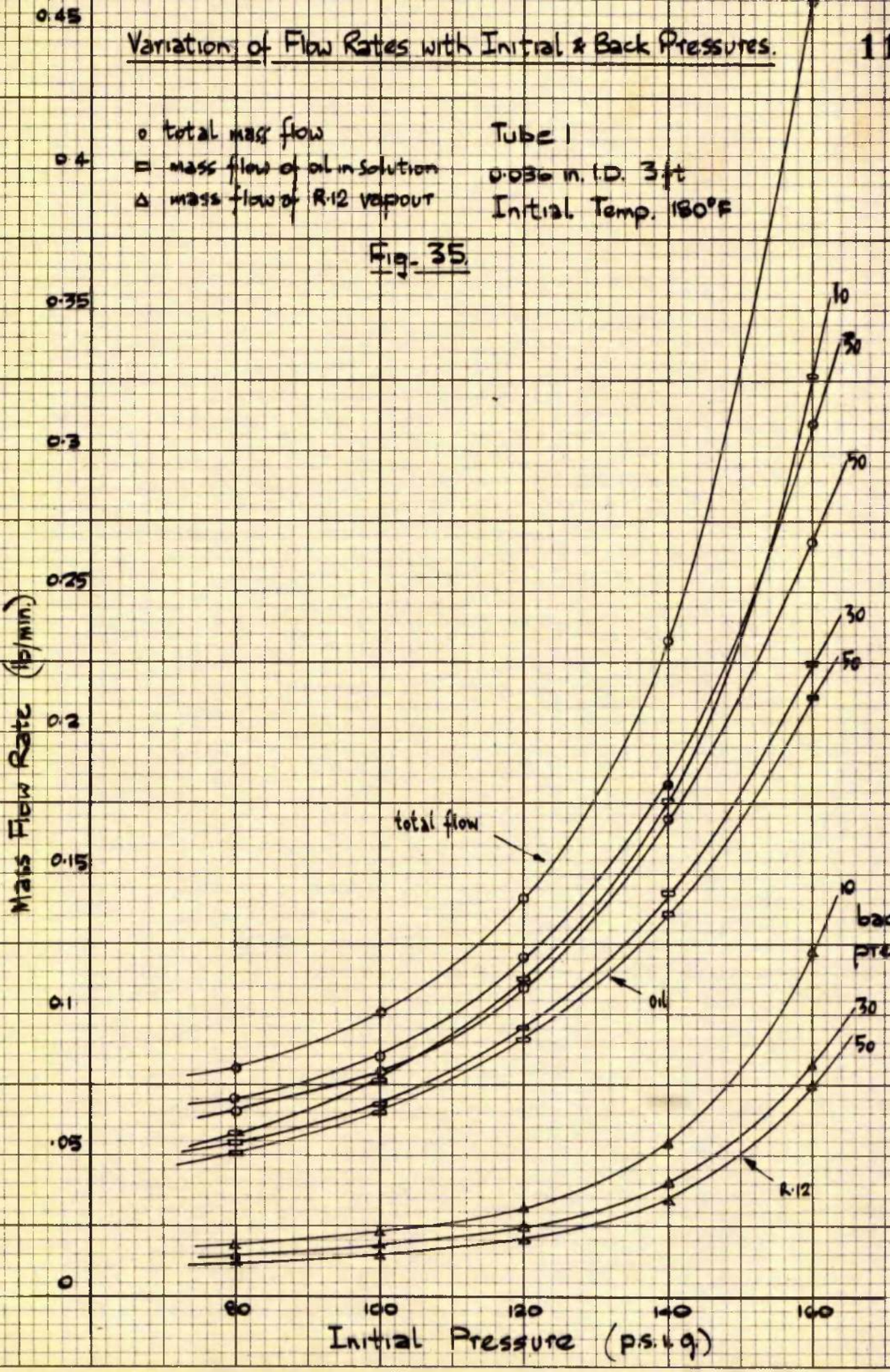
The individual mass flow rates of R12 vapour and of oil under nitrogen pressure were measured also, and are compared with the mass flow rates of oil with R12 in solution, Figs. 45 and 46.

Variation of Flow Rates with Initial & Back Pressures.

o total mass flow
□ mass flow of oil in solution
Δ mass flow of R-12 vapour

Tube I
0.036 in. I.D. 3 ft
Initial Temp. 180°F

Fig. 35.

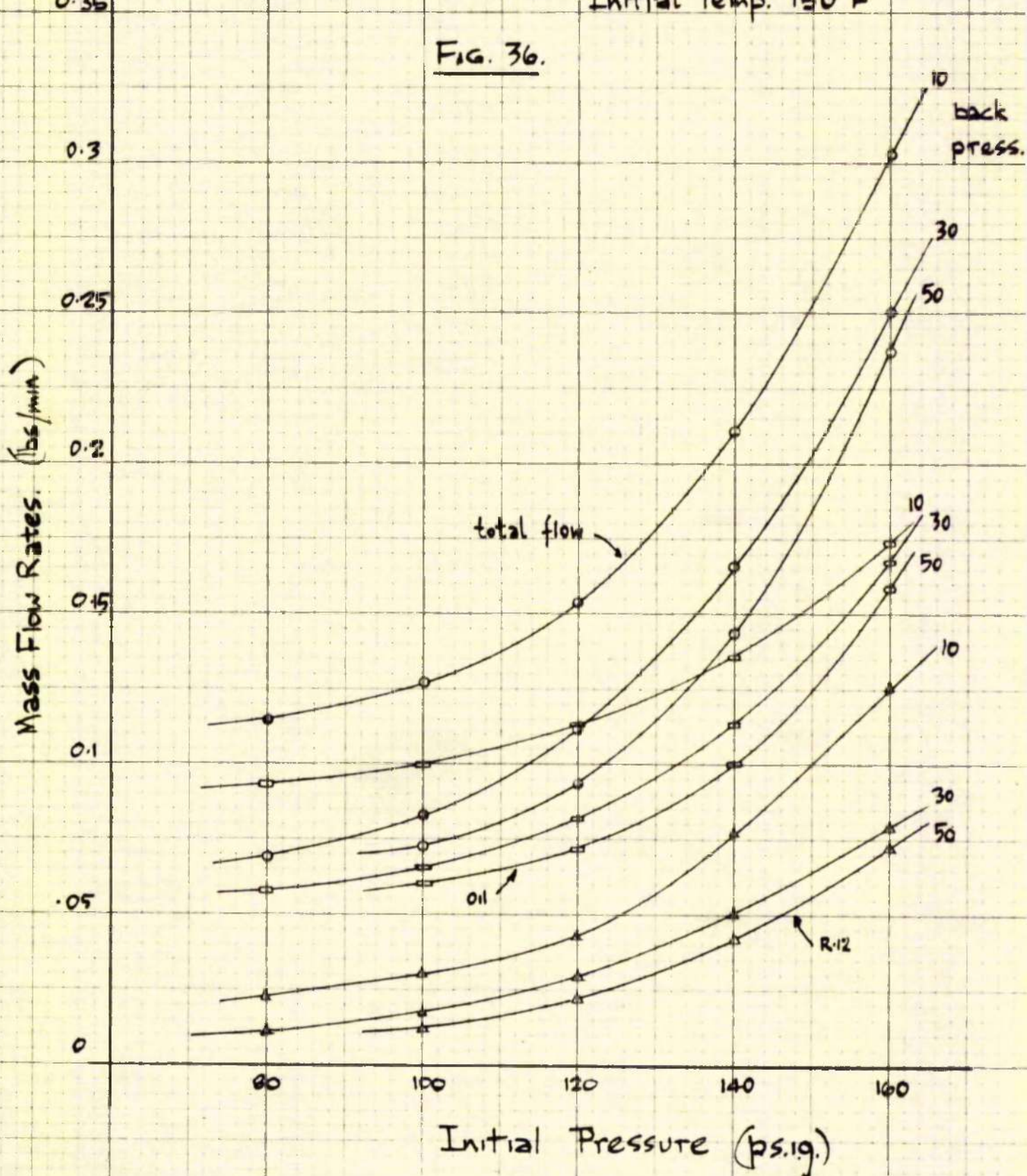


Variation of Flow Rates with Initial & Back Pressures.

o total mass flow
□ mass flow of oil in solution
△ mass flow of R-12 in solution

Tube 1
0.036 in. I.D. 3 ft.
Initial Temp. 150°F

Fig. 36.

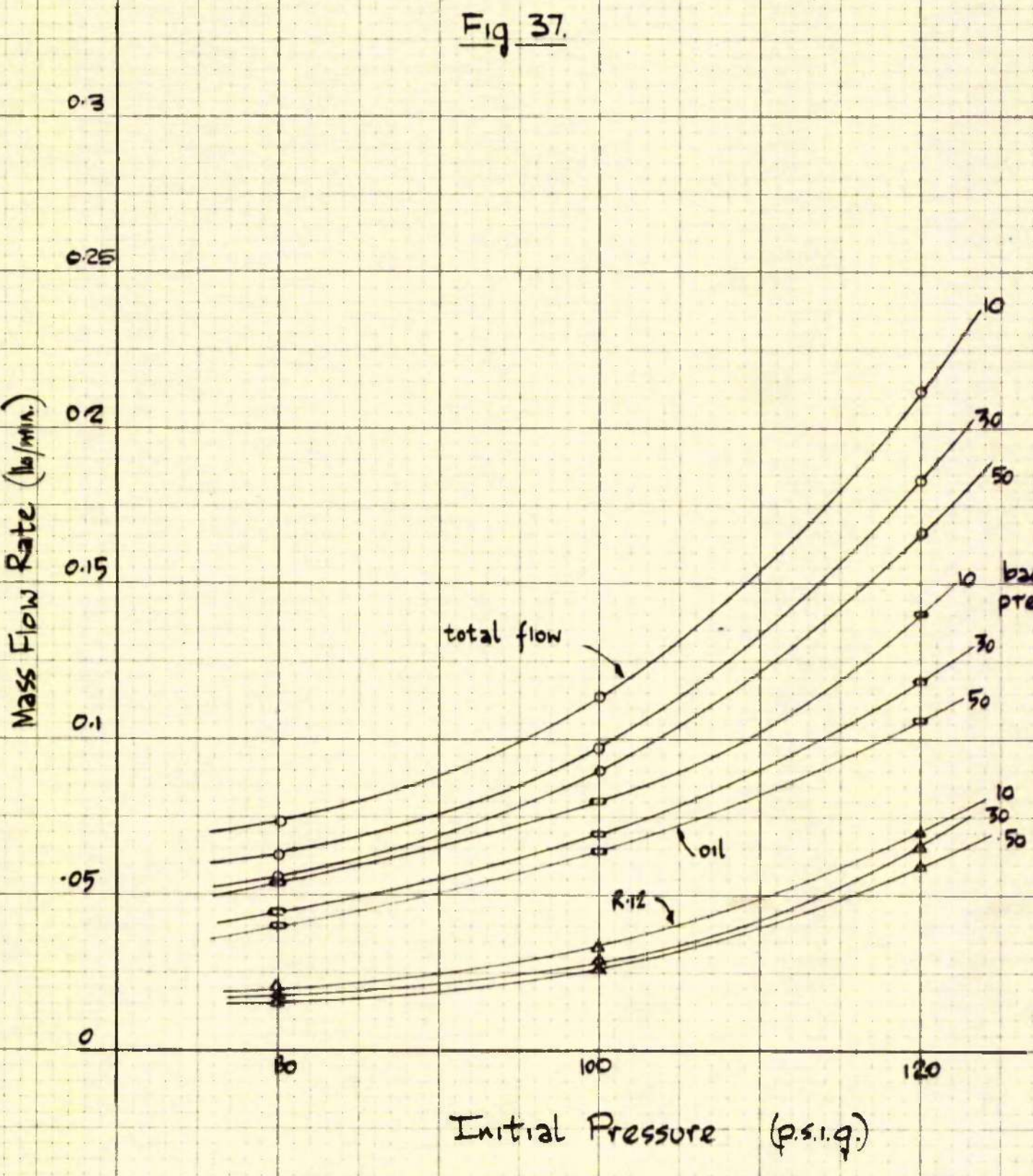


Variation of Flow Rates with Initial & Back Pressures.

o total mass flow
□ mass flow of oil in solution
△ mass flow of R-12 in solution

Tube 1
0.036 in. I.D. 3 ft.
Initial Temp. 120°F.

Fig 37.



Variation of Flow Rates with Initial & Back Pressures.

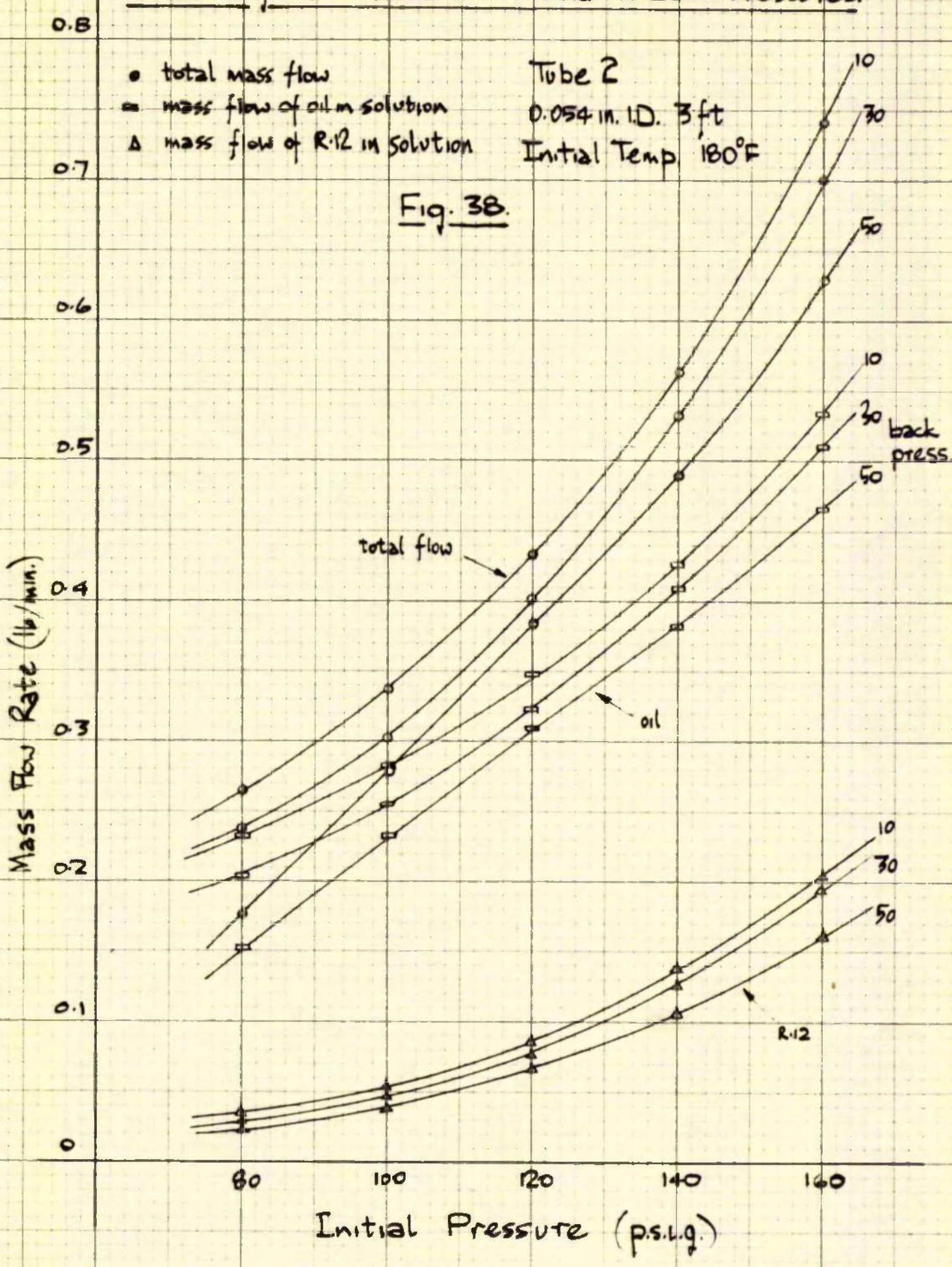


Fig. 38.

Variation of Flow Rate with Initial & Back Pressures.

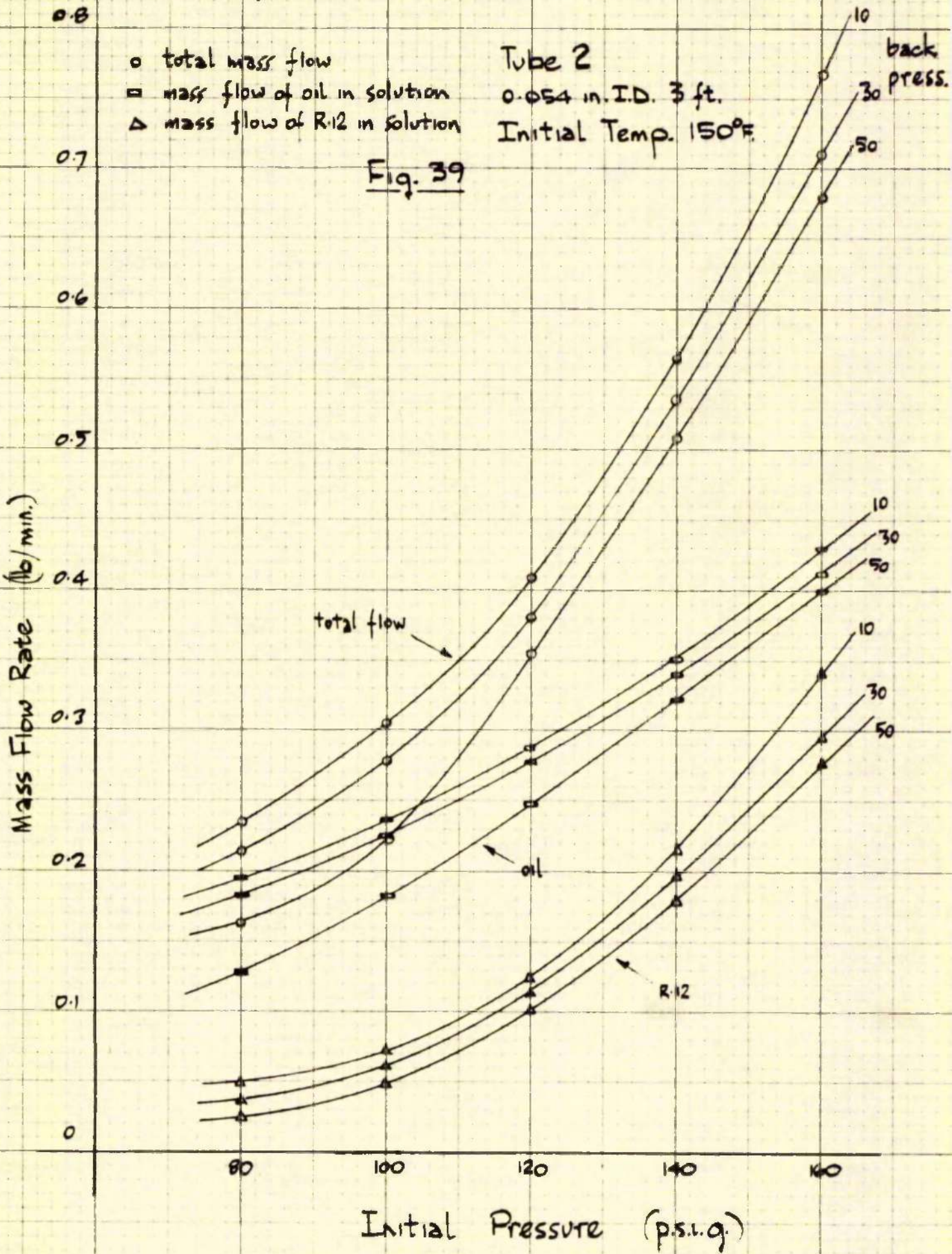


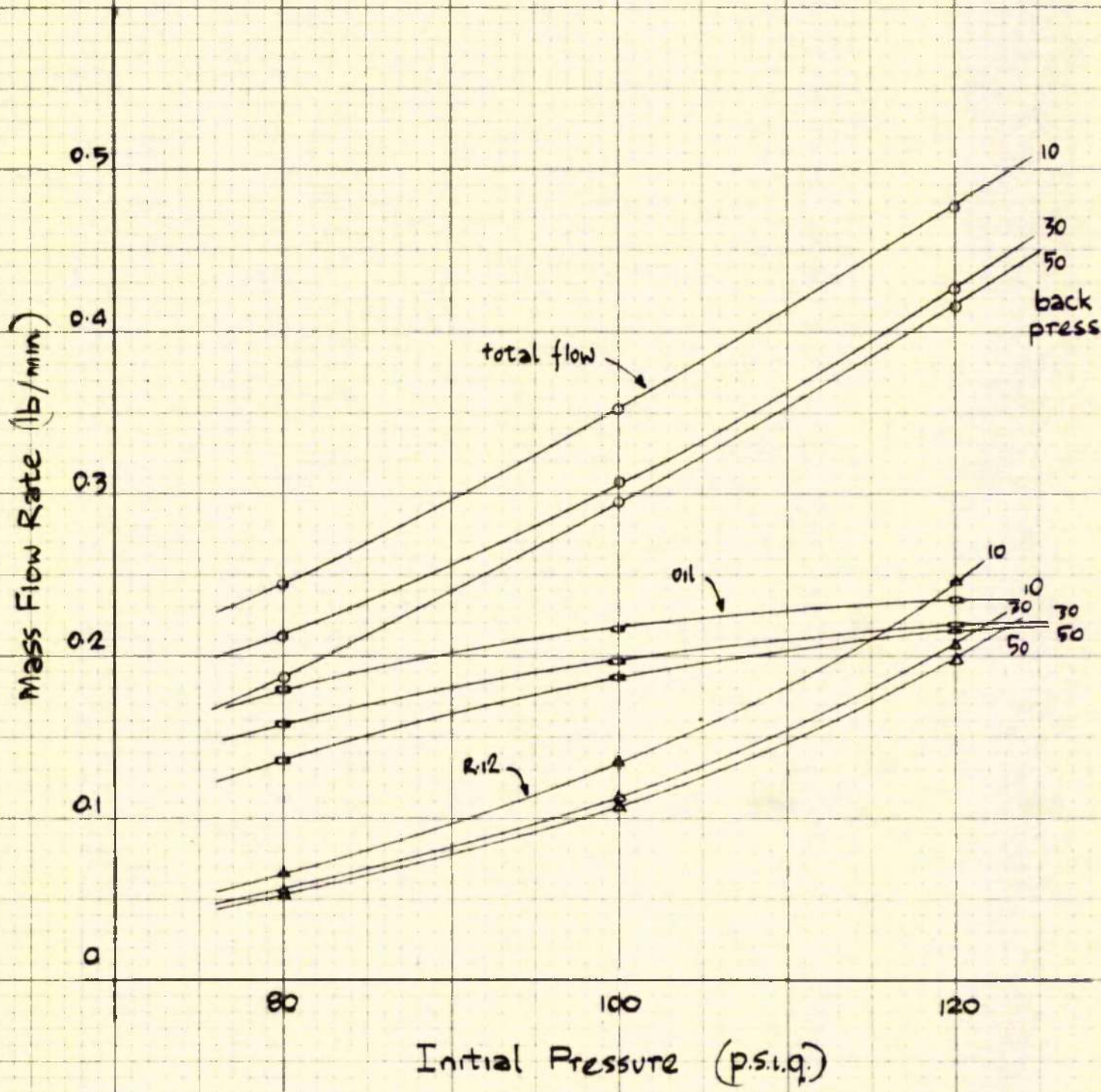
Fig. 39

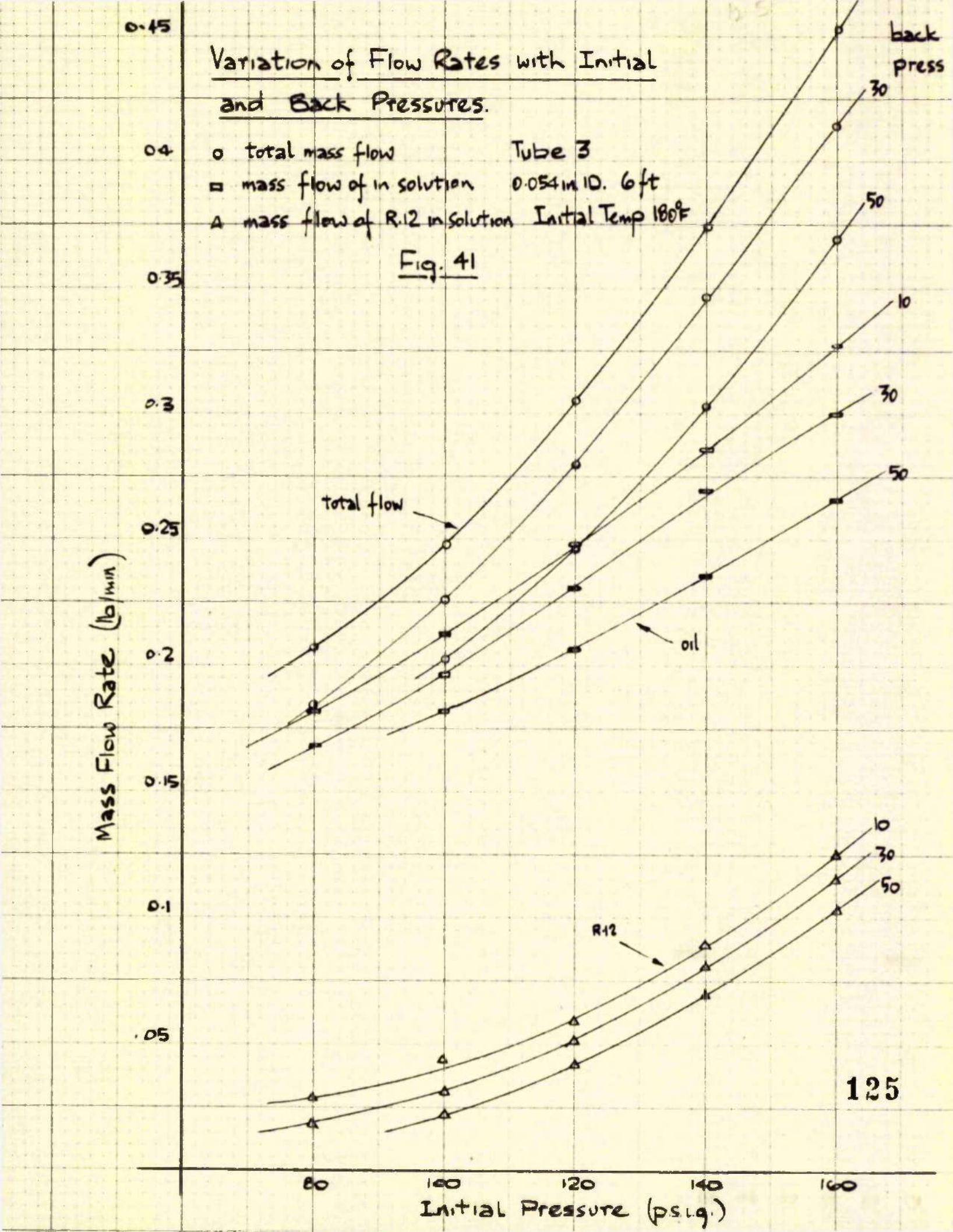
Variation of Flow Rates with Initial & Back Pressures.

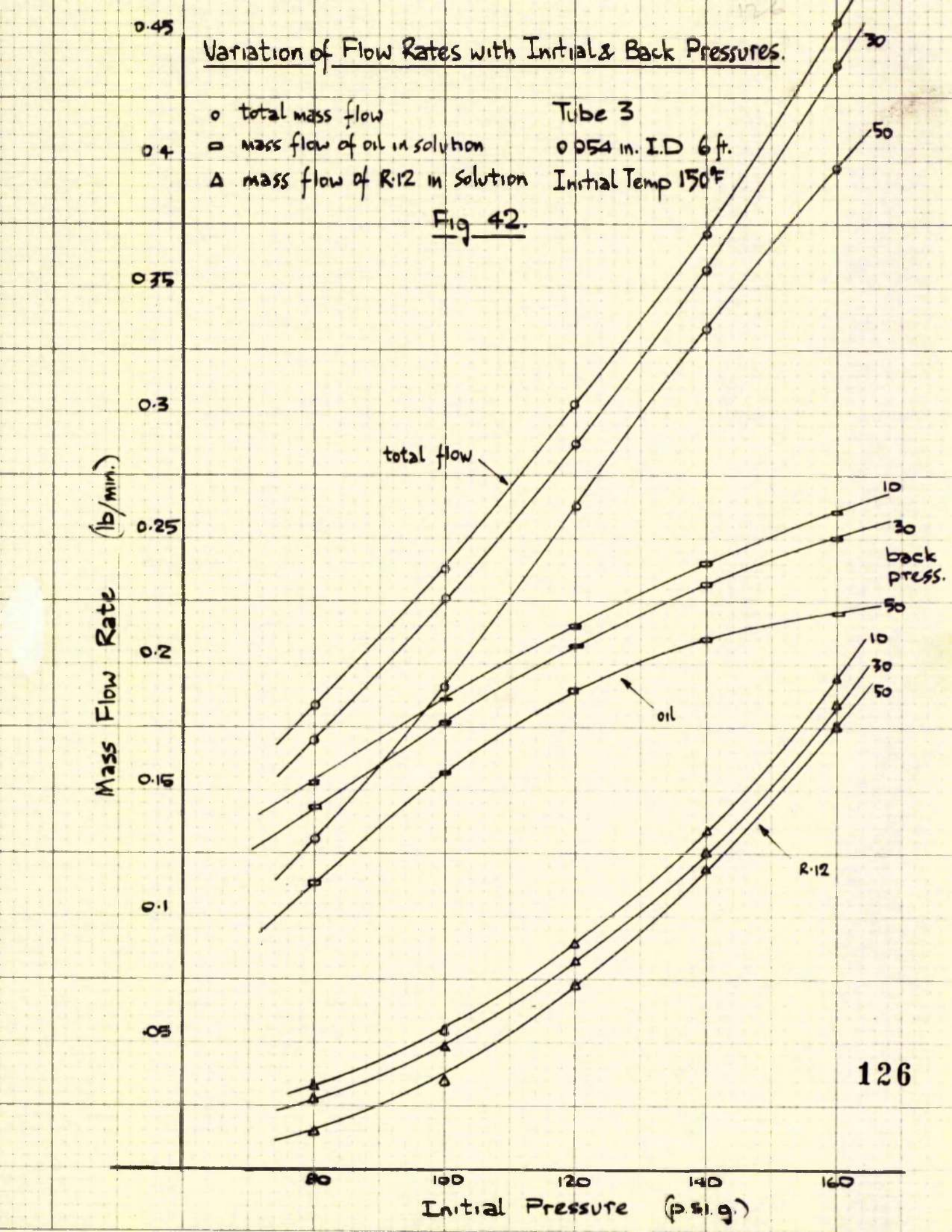
o total mass flow
= mass flow of oil in solution
Δ mass flow of R12 in solution

Tube 2
0.054 in. I.D. 3 ft.
Initial Temp. 120°F.

Fig. 40.




Variation of Flow Rates with Initial and Back Pressures.
o total mass flow
□ mass flow of in solution
Δ mass flow of R.12 in solution
Tube 3
0.054 in I.D. 6 ft
Inital Temp 180°F
Fig. 41
back press
30
50
10
30
50
total flow
oil
10
30
50
R12
Mass Flow Rate (lb/min)
Initial Pressure (p.s.i.g.)
0.45
0.4
0.35
0.3
0.25
0.2
0.15
0.1
.05
80
100
120
140
160

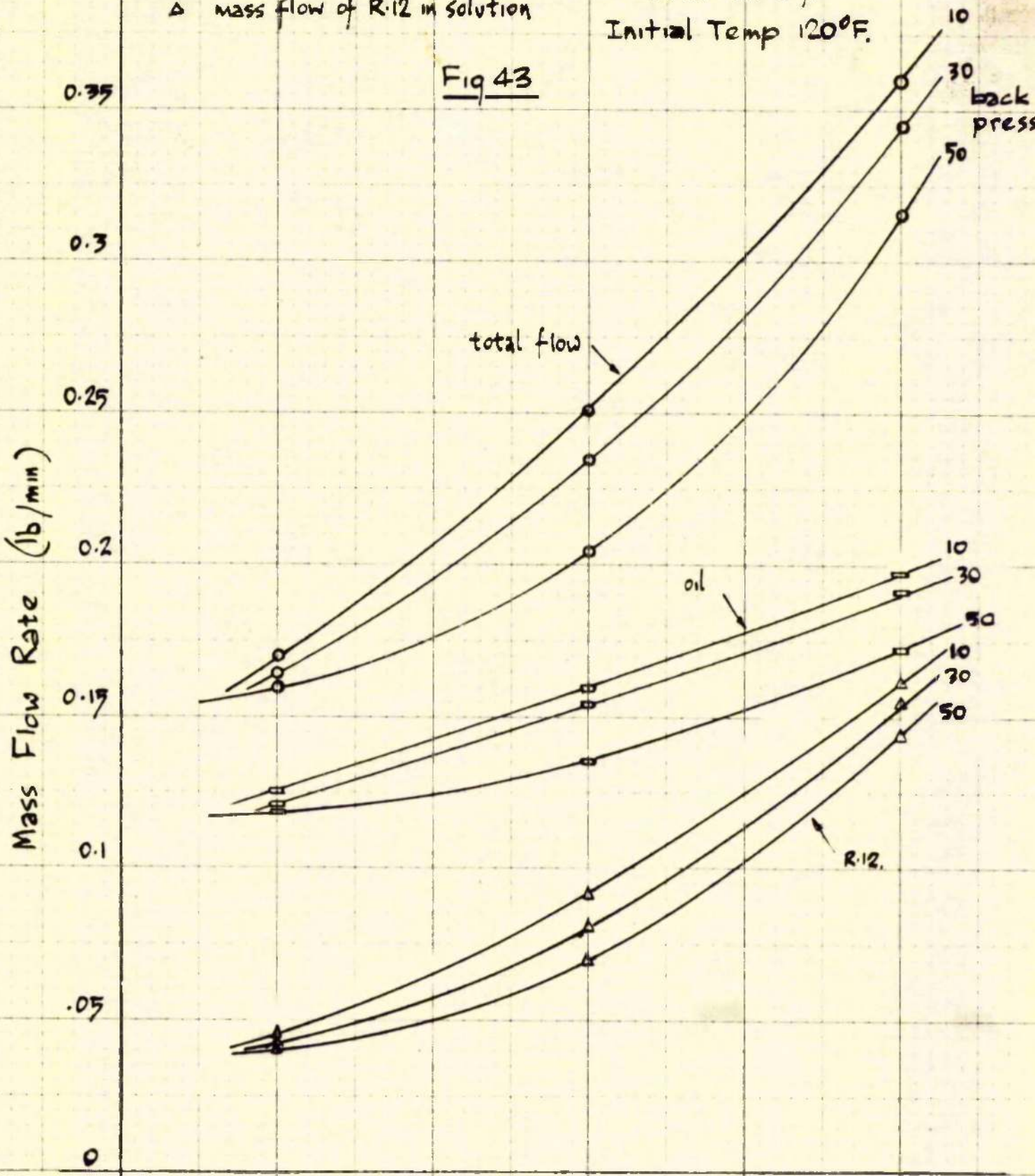

Variation of Flow Rates with Initial & Back Pressures.
o total mass flow
□ mass flow of oil in solution
Δ mass flow of R-12 in solution
Tube 3
0.054 in. I.D 6 ft.
Initial Temp 150°F
Fig 42.
Mass Flow Rate (lb/min.)
0.45
0.4
0.35
0.3
0.25
0.2
0.15
0.1
.05
total flow
oil
R-12
10
30
50
back press.
80
100
120
140
160
Initial Pressure (p.s.i.g.)

Variation of Flow Rates with Initial & Back Pressures

Fig 43

o total mass flow
▬ mass flow of oil in solution
△ mass flow of R-12 in solution

Tube 3
0.054 in. I.D. 6 ft.
Initial Temp 120°F.

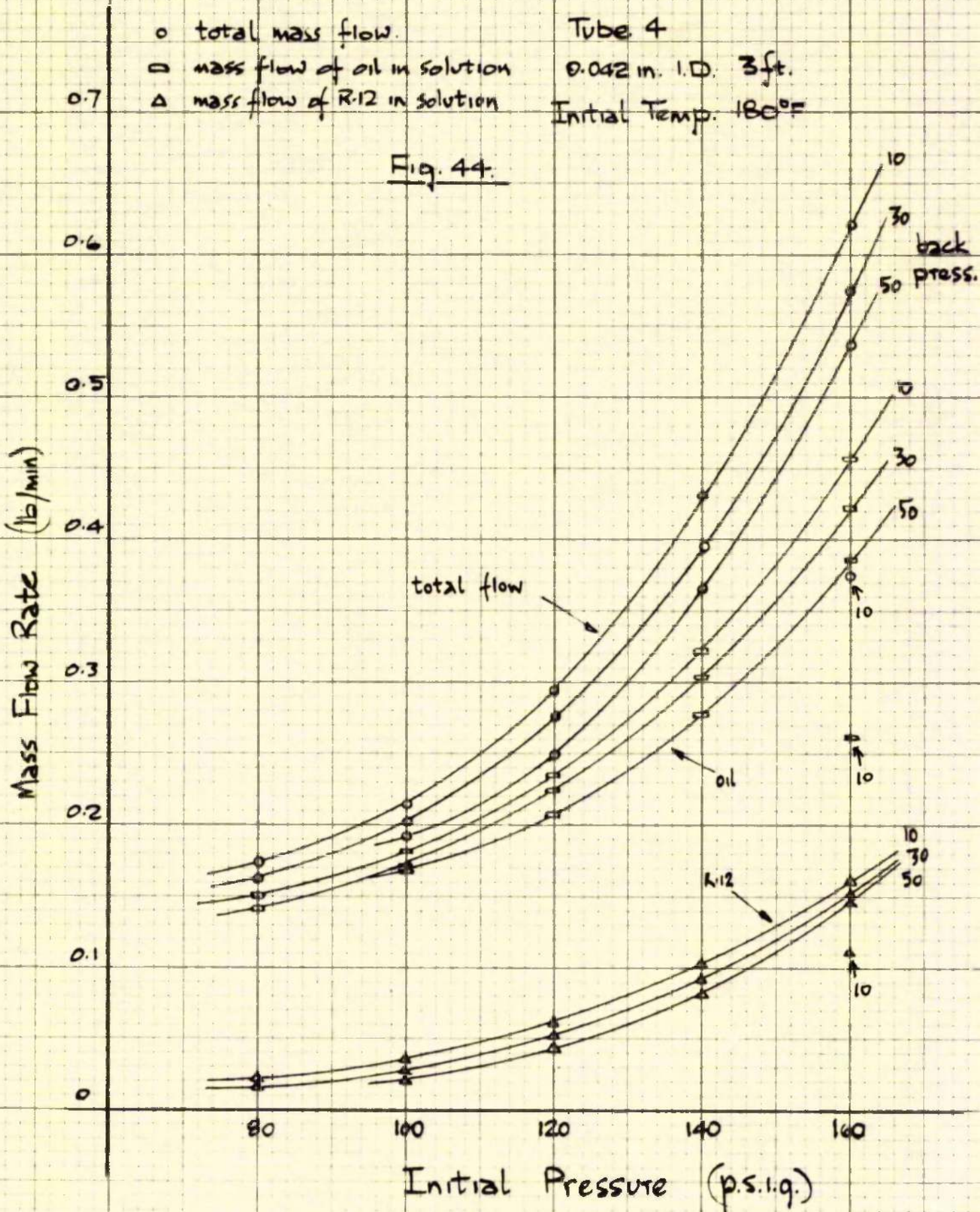


Variation of Flow Rate with Initial & Back Pressures.

o total mass flow.
□ mass flow of oil in solution
△ mass flow of R-12 in solution

Tube 4
0.042 in. I.D. 3ft.
Initial Temp. 180°F

Fig. 44.

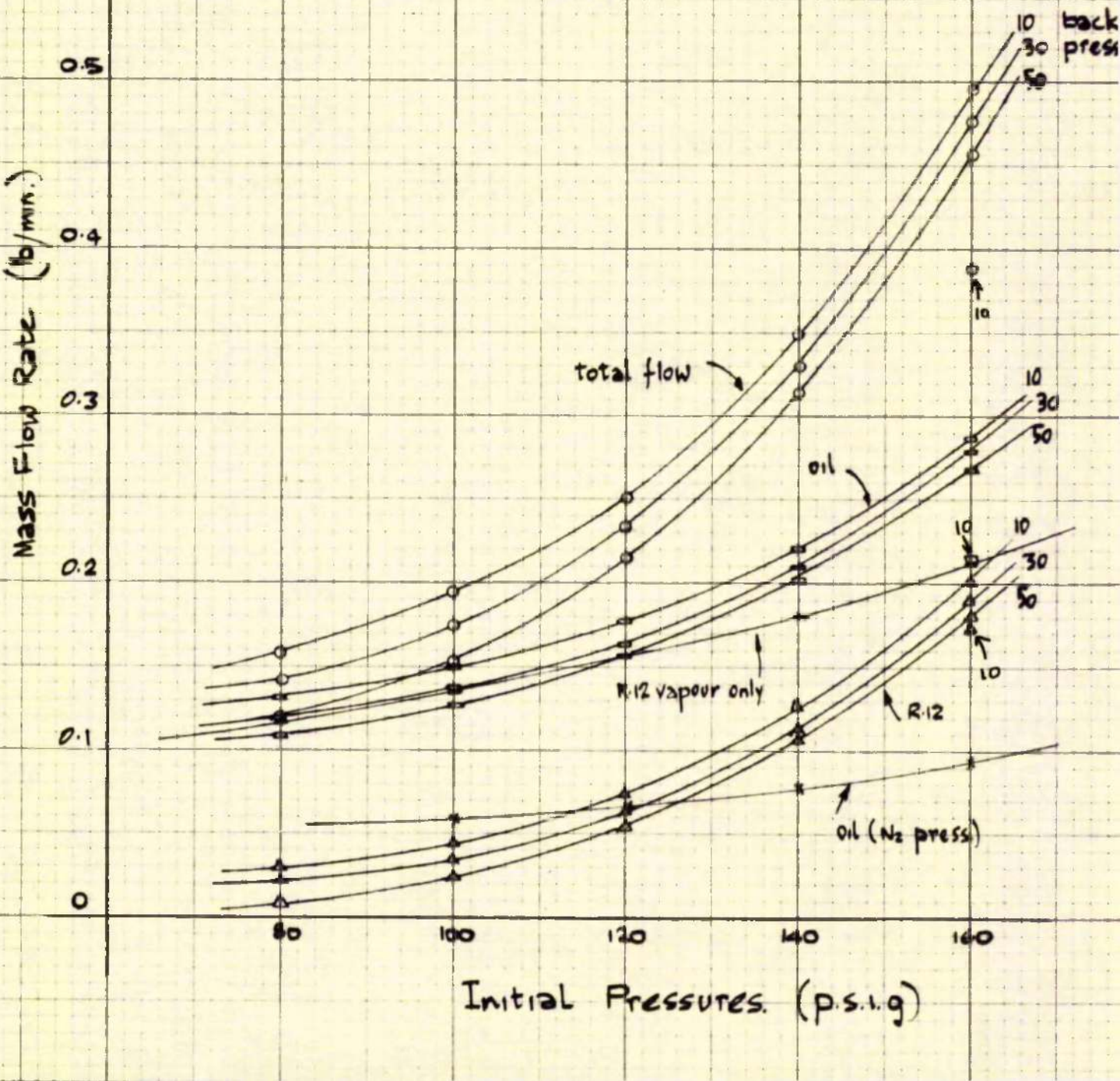


Variation of Flow Rates with Initial & Back Pressures.

- o total mass flow
- ▫ mass flow of oil in solution
- Δ mass flow of R-12 in solution
- x mass flow of oil (N_2 press.)
- - mass flow of R-12 vapour

Tube 4
0.042 in. I.D. 3 ft.
Initial Temp. 150°F

FIG. 45



Variation of Flow Rates with Initial & Back Pressures

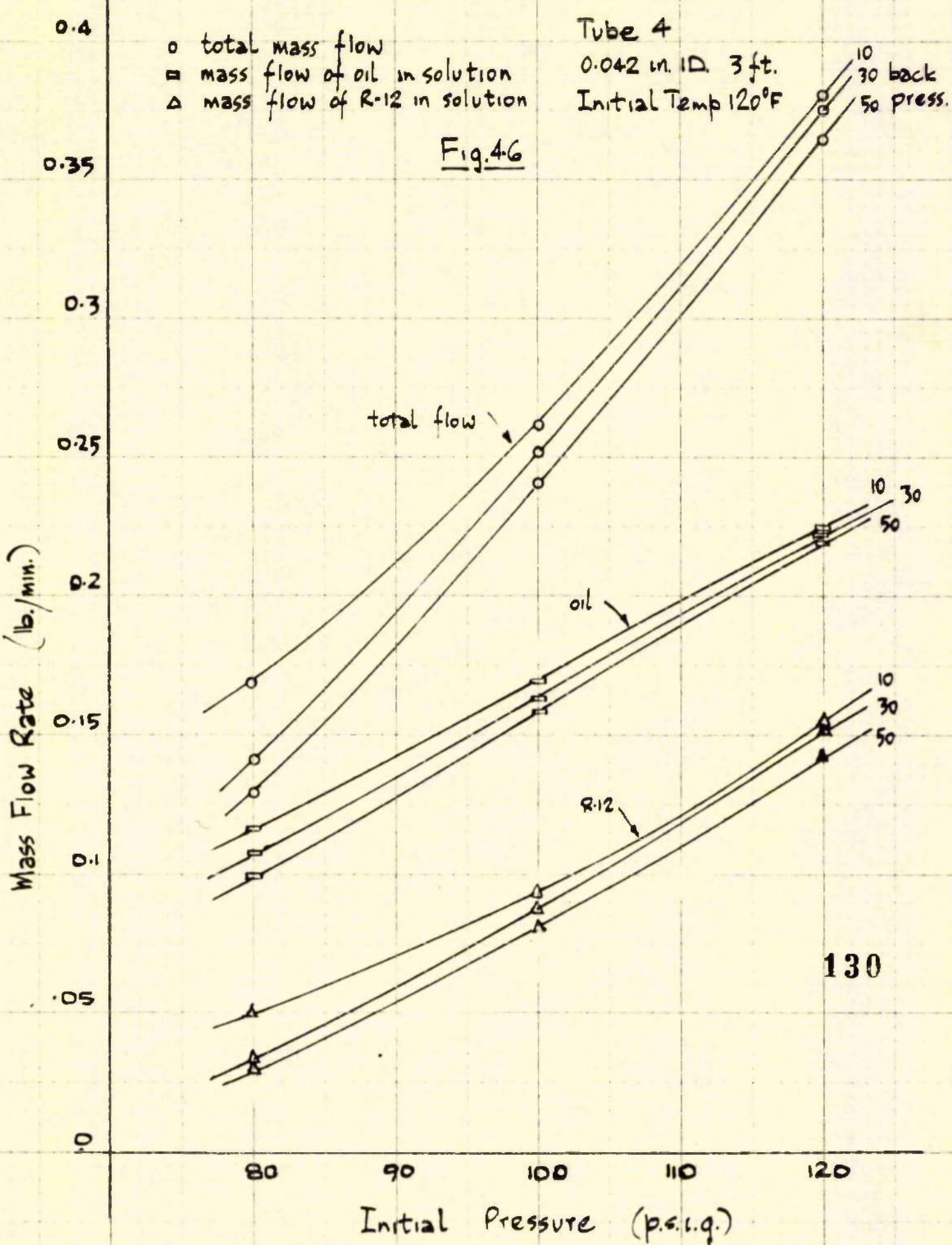


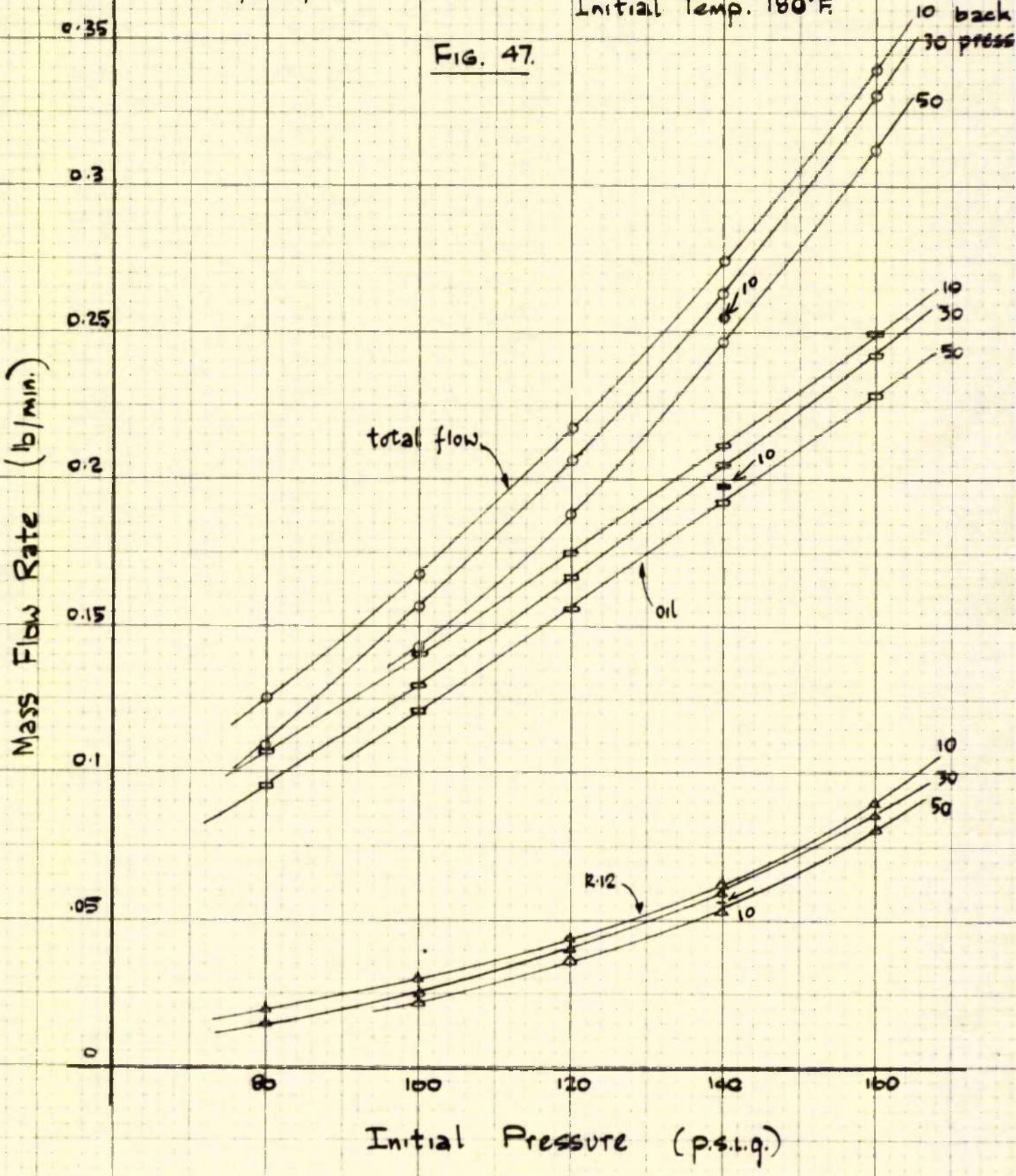
Fig. 4.6

Variation of Flow Rates with Initial & Back Pressures.

o total mass flow
□ mass flow of oil in solution
△ mass flow of R-12 in solution

Tube 5
0.042 in. I.D. 6 ft.
Initial Temp. 180°F.

FIG. 47.



Variation of Flow Rates with Initial & Back Pressures.

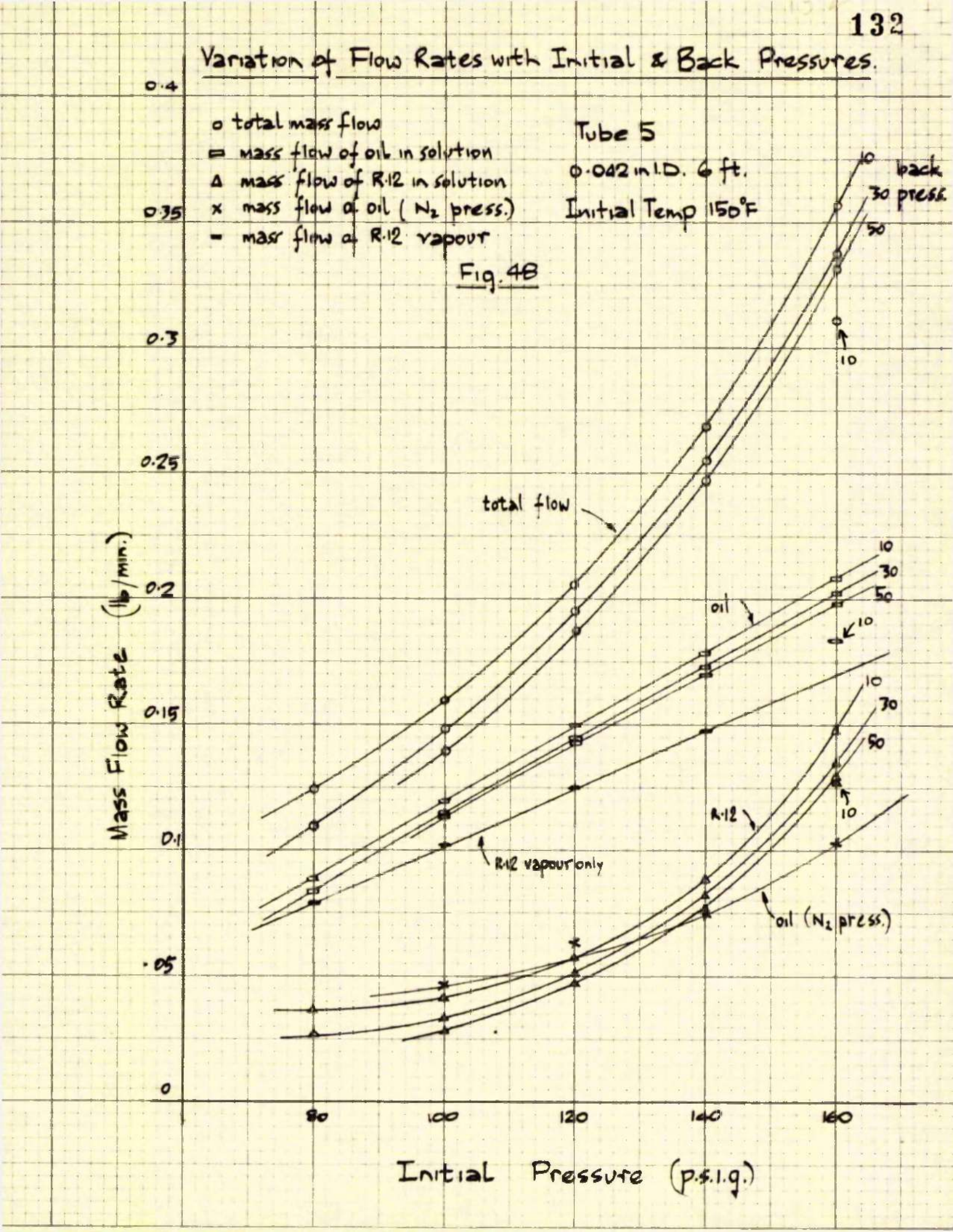


Fig. 48

Variation of Flow Rates with Initial & Back Pressures

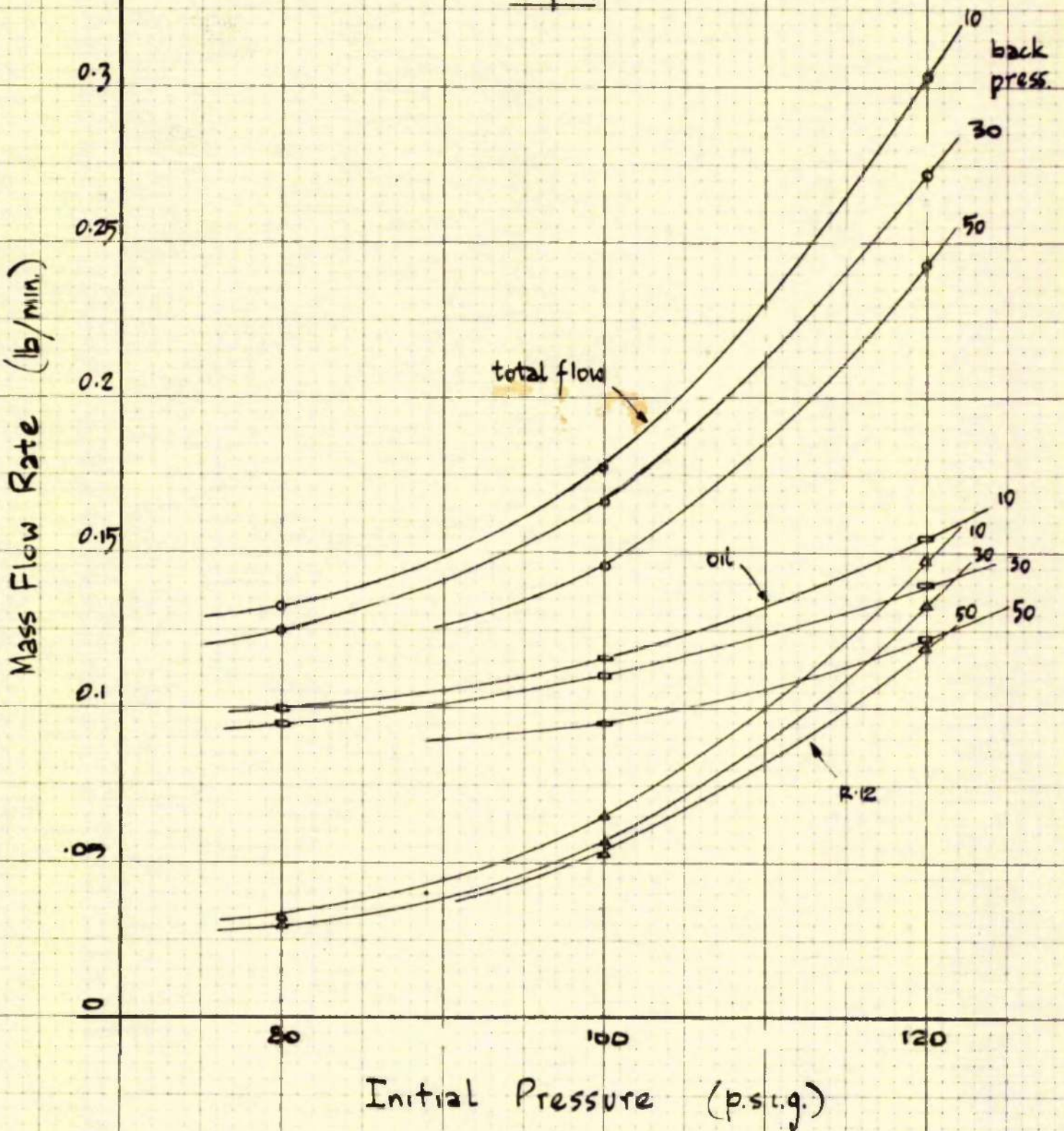
Tube 5

0.042 in. I.D. 6 ft

Initial Temp 120°F.

o total mass flow
□ mass flow of oil in solution
△ mass flow of R-12 in solution

Fig. 49



Chapter 10

ANALYSIS and DISCUSSION

on

RESULTS

10.1 Pressure and Temperature Variation

10.2 Variation of Mass Flow Rate

10.3 Effects of Back Pressure

10.4 Design of Capillary Tube Oil-Return Device

10. ANALYSIS & DISCUSSION ON RESULTS

The results obtained in the final series of tests were more accurate than the results from the preliminary series. Therefore the results presented are for the five capillary tubes tested with the modified apparatus. The accuracy of the experimental flow rates are estimated to be within 15%.

The initial concentration of R12 vapour in the oil-R12 solution varied depending on the pressure and temperature. The range covered in these experiments was from about 50% to 10%. The concentrations obtained from the results are close to the values from the solubility curves; the concentrations measured are compared with the published graph of values, table 6, appendix III.

10.1 Pressure and Temperature Variation

The pressure distribution recorded for different inlet temperatures for flow through tube 2, 0.05" in ID and 3 ft length, and tube 3, 0.05" in ID and 6 ft length are presented in figs. 17 to 25 and 26 to 34 respectively. The pressure and temperature distribution curves for the other tubes were similar

and therefore are not presented.

The distribution curves for 180°F inlet temperature and three back pressures 50, 30 and 10 psig are given in figs. 17 to 19 and figs. 26 to 28, for tubes 2 and 3 respectively. The initial N13 concentration in these cases varies from 29% to 10% approximately.

The pressure distribution for 150°F inlet temperature appear in figs. 20 to 22 and 29 to 31.

The initial N13 concentration in these cases varies from 43% to 15%.

The pressure gradients for both these inlet temperatures are similar, and it is not feasible to predict the beginning of two-phase flow from these curves, or even if two-phase flow occurs.

The N13 concentration was higher for the lower initial temperature of 120°F, and in these cases varied from 50% to 25%, and the pressure distribution curves are presented in figs. 23 to 25 and 32 to 34.

In the flow of these high concentration solutions it is observed the pressure readings near the tube exit were lower than the pattern demonstrated in

the lower concentration flows. For the same initial pressure, and reducing back pressure of 50, 35 and 20 psig, this phenomenon is more prominent. This could be a possible point for the commencement of two-phase flow. There may be a formation of H_2S vapour released from solution, which would cause choking, and a drop in pressure. This tendency would increase with lower back pressure, as observed.

The temperature distribution for $100^{\circ}F$ inlet are presented in fig. 18 and 28 for tubes 2 and 3 respectively. Those for $150^{\circ}F$ and $120^{\circ}F$ inlet temperature are presented in figs. 20 and 29, and figs. 25 and 33 respectively.

It is observed that in all cases, the temperature recorded 2 inches after entrance is lower than that maintained in the receiver. It is also observed that in almost all cases the temperature rises along the tube, and falls again near exit.

The fall in temperature at entrance can be explained by the loss of head at entrance causing a rise in enthalpy of the H_2S. For an adiabatic drop in pressure. A higher enthalpy of H_2S would result in a lower solution temperature, which is observed 2 inches after entrance. Subsequently the lower solution

temperature would cause a lower enthalpy of R12, raising the temperature to a point less than before, as observed. This alternating rise and fall in temperature is a transient condition of very short duration, but it is observed as the fluid is flowing continuously at about 15 to 40 ft/sec in tubes only 3 ft and 6 ft long.

The change in temperature observed is less with lower inlet temperature and shorter tube.

The temperature gradient for pure oil flow, under nitrogen pressure, show an almost isothermal flow, except for a small increase at entrance and a similar small decrease at exit, probably due to the end connection fittings. The temperature gradient for the flow of pure R12 vapour showed an increase in temperature along the tube and falling only near the exit, due again to the end fittings. This can be explained from the fact that frictional loss for R12 vapour is greater than that for oil; the rise in temperature for R12 vapour flow is due to heat generated by frictional losses.

It is possible that the rise in temperature of the solution is due to heat from frictional losses. However, this is unlikely, being in solution R12 is not

in the vapour phase, and it has a lower viscosity. Also the density of solution is greater than the density of oil or R12 vapour and since

$$\text{frictional loss} \propto \frac{16\mu}{\pi D \rho}$$

the higher density and the lower viscosity of the liquid both contribute to reducing the frictional loss, resulting in increased mass flow rates.

10.2 Variation of Mass Flow Rates

The variation of mass flow rates of the oil and R12 components of the solution, and the total mass flow rates of the solution, for different head and back pressures, different inlet temperatures, through tubes of different lengths and bores, are presented in figs. 35 to 49. From these graphs it is observed that the total mass flow rate varies almost as a straight line, with varying head pressures, within the range of 130 to 150 psig.

The flow rate variation curves are almost parallel for the back pressures of 30, 20 and 10 psig.

Considering mass flow rate as a straight line function of head pressure,

$$M_s = P_h \tan\beta + C$$

where $\tan \beta = K(t)^a \frac{(d)^b}{(L)^c}$

substituting the constants, a, b, c, C and K from experimental results

$$\tan \beta = 0.000369\ (180 + \Delta t)^{0.69}\ \frac{(d)^{0.6}}{(L)^{0.55}}$$

and $C = 0.0024(\Delta p_1) + \frac{2.80}{(L)^{0.55}(d)^{1.25}(d-20)^{1.55}} - \frac{(p_b - 20)}{(36\ p_b)}$

where W_s = mass flow rate of oil-HF solution, lb/min

p_h = head pressure, psig

Δp_1 = $(p_h - 150)$ if $p_h > 150$ psig, or

= $(110 - p_h)$ if $p_h < 110$ psig

p_b = back pressure, psig

t = initial temperature of solution, °F

Δt = $(180 - t)$

d = internal diameter of capillary tube, inches

L = length of capillary tube, ft.

Using this equation the mass flow of oil-HF solution can be calculated for different head pressure, and inlet temperature for a tube of known diameter and length. The variation of flow rates deviates from

the straight line outwith the head pressure range of 110 to 150 psig. A correction factor is therefore introduced in the value of the constant C, when the pressure is below 110 psig or higher than 150 psig.

Values calculated using this equation are estimated to be within 10% of the results presented.

Although the total mass flow varies appreciably with changes in inlet temperature, the oil component of the flow rate is not affected greatly; the change occurs chiefly in the R12 component of the flow.

The R12 dissolved in oil reduces the viscosity of the oil, depending on the concentration. The concentration, for the same pressure, decreases with increase in temperature. Therefore, the viscosity of the solution tends to increase with increase in temperature, while the oil component gives a counterbalancing effect, as an increase in temperature causes a decrease in the oil viscosity.

Comparison between the flow rate of pure oil under nitrogen pressure and the oil component of the oil-R12 solution, in figs. 49 and 48, for tubes 4 and 5 respectively, shows an increase of about 100 to 300%

in the flow rate of oil when it is a component in solution, mainly due to the reduction of viscosity by the presence of R12 in solution.

10.3 The Effect of Back Pressure

In general increasing back pressure reduced the mass flow rate. Three back pressures were used in the tests, 10, 30 and 50 psig. The reduction in mass flow when the pressure was raised from 30 to 50 psig, was less than the reduction in flow when the pressure was raised from 10 to 30 psig.

Bolstad and Jordan (5) when investigating flow of liquid R12 through the expansion capillary observed that variations in the back pressure did not affect the mass flow rate to any appreciable extent. This does not agree with the present observations on the flow of oil-R12 solutions through the oil-return capillary.

In some cases two different mass flow rates were obtained for the same experimental conditions. Such points are indicated in figs. 44, 45, 47 and 48. They were invariably for the lower back pressures, and for the higher head pressures of 160 and 140 psig. While one set of readings are consistent with the

general pattern of flow, the second set of readings give considerably lower mass flow rates.

Lower back pressure, in addition to providing a greater pressure difference for the flow, also increased the possibility of R12 vapour separating from the solution, because of even lower solubility. Any R12 vapour formed and the remaining solution would then be a two-phase frothing flow. The R12 vapour being of much lower density, would account for the reduction in flow rate.

The oil-R12 solution is reasonably stable, and super-saturation may occur, permitting liquid flow, as observed in most cases where the points fall on smooth curves over the whole range of pressure differences. However, if there is a disturbance to restore equilibrium in the super-saturated solution, vapour will be formed, and two-phase flow will occur, as observed, for some of the tests at 10 psig back pressure, figs. 44, 45, 47 and 48.

The two-phase flow of oil-R12 solution is non-evaporating, and therefore there is no indication of cooling due to evaporation; hence the temperature distribution curves do not give any indication whether or where two-phase flow commences.

10.4 Design of Capillary Tube Oil-Return Device

A design example is given in Appendix II.

The oil in the separator is considered to be a saturated solution of oil and R12, since excess R12 is always present in the separator.

There is a leakage of R12 vapour, when the capillary entrance is not flooded. It is shown that this leakage is small and permissible under actual operation.

The theoretical loss in refrigeration capacity at the worst condition, due to the oil-return capillary running dry, is estimated. In practice the loss will be further reduced because (1) oil pumped over will not be as low as half the design capacity, (2) the R12 vapour will not flow separately; there will be intermittent flow of R12 vapour and oil solution, resulting in two-phase flow. The capacity of the capillary for two-phase flow is considerably lower than for single phase flow of solution, (3) the oil pumped over is low when the discharge pressure falls; this will reduce the capacity of the capillary in two ways, (a) due to lower head pressure, and (b) due to lower solubility of R12, the viscosity

of the solution will increase, reducing the flow rate, (b) the capillary entrance may be so arranged that the least quantity of oil solution is sufficient to flood the capillary entrance.

An additional advantage of the oil-return capillary tube is that, being an open restriction, the smallest increase in the oil level in the separator will suffice to flood the capillary entrance and resume the flow back.

In systems using the valve mechanism, the ball or float is actuated by the oil level in the separator, therefore the oil must attain a minimum level before the valve opens to allow the return flow to commence.

This advantage of the open capillary tube offsets the small loss in refrigerating capacity. The oil-return capillary also offers the advantages of reduced cost, simplicity in construction and trouble-free operation.

Chapter 11

CONCLUSIONS

11. CONCLUSIONS

1) The mass flow rate of oil through a capillary tube is increased by R12 in solution, due mainly to lower viscosity. Except at high initial concentrations and large pressure differences with low back pressures, the flow appears to be all liquid.

2) With high concentrations at high initial pressure and low back pressure, the flow may be unstable. Non-evaporating two-phase flow may occur; at other times under the same conditions, liquid flow occurs, suggesting that the solution is super-saturated.

3) The mass flow rate is not a function of pressure difference only, but also of the initial pressure, because on the latter depends the initial R12 concentration. For a particular pressure difference, the variation in mass flow for a change in back pressure is small compared to that obtained with a similar difference in initial pressure.

4) Although the mass flow of solution varies considerably with initial temperature, the oil component of the flow is almost constant. Higher initial temperature reduces the oil viscosity and also lowers the R12 solubility; lower solubility increases

the viscosity of the solution. These two effects counteract each other, resulting in the almost constant flow rate of the oil component of solution.

5) Since the oil-return capillary is an open construction, there is a leakage of NH3 vapour, when the capillary entrance is not flooded. This leakage is only one to two times the weight of NH3 leaked in solution when flooded, and a negligible reduction in refrigerating capacity results.

6) It is not necessary to design for the maximum oil pumped over condition to obtain satisfactory performance.

7) A small increase in the oil level is sufficient to flood the capillary entrance. Hence the return flow may be continuous unlike the ball or float mechanism now in use.

8) The mass flow of oil-NH3 solution can be computed approximately from the empirical relation

$$W_s = P_h \tan \beta + 0$$

where W_s = mass flow of solution, lb/min

P_h = head pressure, psig

$$\tan\beta = 0.000585\ (180 + \Delta t)^{0.89}\ \frac{(d)^{0.6}}{(L)^{0.66}}$$

$$Q = 0.0024(\Delta p_1) - \frac{7.82}{(L)^{0.85}(d)^{1.85}(t - 90)^{1.53}} - \frac{(P_b - 10)}{(34\ P_b)}$$

Δp_1 = $(P_h - 150)$ if $P_h > 150$ psig, or

= $(110 - P_h)$ if $P_h < 110$ psig

P_b = back pressure, psig

t = inlet temperature, °F

Δt = $(180 - t)$

d = internal diameter of tube, inches

L = length of tube, feet

9) The capillary tube can be used, and may now be designed from the comprehensive results presented, as an oil-return device from the oil separator to the compressor crank-case, providing a simple, trouble free and economic substitute to the valve mechanism in general use at present.

10) Capillary tube dimensions for evaporating two-phase flow of refrigerant, as in the expansion capillary, or single phase or non-evaporating two-phase flow of oil-refrigerant solutions, as in the oil-return capillary, must of necessity be obtained by empirical

or semi-empirical design. Hence verification and development will be by experiment only. Advance will depend on adequate facilities to provide accurate evidence. The experimental plant described is the first developed to investigate the phenomena of oil-air solution flow.

Chapter 12

SUMMARY AND TEST

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APPENDICES

APPENDIX I

I.1 Sample Calculation of Test Results

Test No. 112

Measurements in

(1) Oil Recovery

		Final	Initial
Press	(psia)	20.4	24.1
Temp	(^{o}F)	140	139.3
V_o =	(cc)	839	42

From oil-N2 solubility chart, Fig. 50. percentage oil by weight is

% oil	96	95

From density-concentration-temperature relation for oil-N2 solution, Fig. 52 .

Density	(lb/cc)	1.927	1.940
	(x 10^3)		

∴ weight of oil solution

(lb)	1.553	0.079

Nett quantity of oil collected V_o = 1.472 lb

Weight of N2 in solution

(lb)	0.0650	0.0051

Nett weight of N2 in solution collected V_{N2} 0.0599 lbs

Volume of R12 vapour in the oil receiver =

total volume of oil receiver, 240) ce less volume of solution collected

V_{x2}	(cc)	564	1961

Specific volume of R12 vapour, at respective pressures and temperatures, from pressure-enthalpy chart for R12

	cc/lb	70799	50466

∴ weight of R12 vapour collected

	lb	0.0080	0.0389

∴ nett weight of R12 vapour finally in the oil receiver W_{x2} = 0.0192 lb

(Note - this quantity of R12 is to be subtracted from the total, as the weight of R12 vapour in the oil receiver at initial conditions is greater than that at the final conditions).

(3) R12 Receivers

receivers in use, nos. 1 and 2

Press	(psia)	16.7	1.97
Temp	(^{o}F)	70.2	68.7
volume	(ft^3)	1.599	1.599

Specific volume of R12 vapour, at respective conditions of pressure and temperature, from pressure-enthalpy chart for R12

(ft^3/lb)	2.69	23.8

∴ weight of R12 vapour collected in the R12 receivers

0.5923	0.0607

∴ nett weight of R12 collected V_{x3} = 0.5235 lb

∴ nett total weight of R12 collected during test

V_x = 0.5625 lb

and nett total weight of oil collected during test

V_o = 2.0420 lbs

Duration of test = 2.75 minutes

∴ flow rate of solution, W_s = 0.7425 lb/min

flow rate of oil component, W_o = 0.5394 lb/min

flow rate of R12 component, W_x = 0.2031 lb/min

Percentage of R12 concentration of solution in H.P. receivers, at the commencement of test, from measured quantities = $\frac{0.2021}{0.7425} \times 100$ = 27.69 %

From solubility chart, Fig.50, R12 concentration is

= 28 %

(The percentage concentration in other cases agrees to within about ½% see table 6, appendix XX.)

X.2 Calculation for R12 Vapour Flow

The equation used to calculate the flow rate of R12 vapour through capillary tubes was taken from

Hopkins (13) relation for air flow.

$$\text{Air flow } (ft^3/min) = K \frac{(p + 2.3)^{0.94} d^{3.6}}{l^{0.505}} \quad (I.1)$$

where K is a constant

p = pressure difference in psi

d = inside diameter of tube, inches

l = length of tube, feet

the value of K used is 3.235, for R12 vapour flow for

tube 4 d = 0.042 in

l = 3 ft

for head pressure of 140 psig and back pressure of 10 psig p = 130

$$\therefore \text{R12 flow } (ft^3/min) = 3.235 \frac{(130 + 2.3)^{0.94} (0.042)^{3.6}}{(3)^{0.505}}$$

$$= 0.0484 \text{ ft}^3/\text{min}$$

From pressure-enthalpy chart for R12 specific volume at 140 psig and 150°F

$$= 0.2955 \text{ ft}^3/\text{lb}$$

∴ mass flow rate of R12

$$W_{ct} = \frac{0.0484}{0.2955} = \underline{0.1640} \text{ lb/min}$$

actual measured flow rate

$$W_{ca} = \underline{0.182} \text{ lb/min}$$

X.3 Calculation for Oil Flow

For liquid flow head lost due to friction

$$F_h = \frac{4 f L v^2}{2g D} \text{ ft}$$

neglecting other losses, frictional loss equals total head lost

$$\frac{\Delta p}{w} = \frac{4 f L v^2}{2g D}$$

Reynolds number for oil flow, using experimental mass flow, for tube 4

ID of tube = 0.042 in

length = 3 ft

W_o = 0.091 lb/min at 160 psig head pressure and 150°F inlet temperature

also $m = A v \rho$

or $\frac{m}{A} = v\rho = \frac{0.091 \times 144}{\frac{\pi}{4}(0.042)^2 \times 60}$

$\therefore \quad v\rho = 157.7 \frac{ft}{sec} \times \frac{lb}{ft^3}$

Reynolds number $R_o = \frac{v D \rho}{\mu} = \frac{157.7 \times 0.0035}{9.6103 \times 10^{-4}}$

$= \underline{574.0}$

$R_o = 574 < 2000$, therefore, laminar flow friction factor is used

$$f = \frac{16}{R_0} = \frac{16\mu}{VD\rho} = \frac{16\nu}{VD} \qquad (I.3)$$

substituting in equation (I.2)

$$\frac{\Delta p}{W} = \frac{4 L V^2}{2g D} \frac{16\nu}{VD}$$

rearranging

$$V = \frac{\Delta p}{W} \frac{2g D^2}{64 L \nu} \qquad (I.4)$$

now mass flow $M = A V W$

substituting the value of V from equation (I.4)

$$M = A W \frac{\Delta p}{W} \frac{2gD^2}{64 L \nu}$$

$$= \frac{\Delta p \pi g D^4}{4 \times 32 L \nu}$$

if Δp is given in lb/in^2

d inches

L feet

ν centistokes

$g = 32.2\ ft/sec^2$

the relation for M, lb/min, reduces to

$$M(lb/min) = \frac{60}{144} \frac{\Delta p \pi g d^4}{4 \times 32 L \nu}$$

$$= 0.3295 \frac{\Delta p d^4}{L \nu}$$

for tube 4, 0.042 in ID and 3 ft length

$$M = 0.3295 \frac{\Delta p}{\nu} \frac{(0.042)^4}{3}$$

$$= 3.415 \times 10^{-7} \frac{\Delta p}{\nu}$$

I.4 Calculation for Reynolds number for the flow of oil-R12 solution, using experimental mass flow

$$m = A v \rho$$

using mass flow for tube 4 at 160 psig head pressure, 15°F inlet temperature, and 10 psig back pressure

or $$\frac{m}{A} = v\rho = \frac{0.404 \times 144}{60 \times \frac{\pi}{4} \times (0.042)^2}$$

$$= 856 \frac{ft}{sec} \times \frac{lb}{ft^3}$$

$$R_e = \frac{V D \rho}{\mu} = \frac{0.042}{12} \times \frac{856}{1.67 \times 10^{-3}}$$

$$= \underline{1794.0}$$

APPENDIX II

Design Calculation

Problem. An oil separator is used on a refrigerant compressor, discharging at 140 psig, 170°F, and using R12 evaporating at 30°F, the capacity is 2400 Btu/hr. The system operates at 95% load for 40 minutes every hour, at 110% load for 4 minutes, 80% load for 6 minutes, 70% load for 5 minutes and is off cycle for 5 minutes every hour. The oil used is 300 SSU, naphthenic base oil. A capillary tube is to return the oil from the separator to the compressor. Standard 0.042 in ID tube is available. What length of tube is required?

Solution:

Considering 100% load, evaporating at 30°F, the R12 charge in the system should be

$$\frac{400}{65.361 \times 0.96} = 6.37 \text{ lb/min}$$

where the latent heat of evaporation at 30°F = 65.361 Btu/lb, and the suction gas has a dryness fraction of 0.96.

Experience indicates that the quantity of oil pumped over is 4 to 7% of R12 by weight. Assuming

at 95% load, oil pumped over is 6%

∴ weight of oil pumped over

$$M_o = 0.06 \times 6.37 = 0.382 \text{ lb/min}$$

This will be design condition for the oil-return capillary tube.

At 140 psig discharge, and temperature 170°F the R12 solubility in oil is 27%, from Fig. 50

∴ weight of solution to be passed by the oil-return capillary tube

$$M_s = \frac{0.382}{0.73} = 0.523 \text{ lb/min}$$

(Note - It can be assumed that initially the oil is saturated with R12, at the discharge pressure and temperature, being in the presence of excess R12)

Using the empirical relation deduced from the experimental results obtained,

$$M_s = P_h \tan\beta + C$$

where M_s = weight of oil-R12 solution, lb/min

P_h = head pressure, psig

$$\tan\beta = 0.000585(180 + \Delta t)^{0.89} \frac{(d)^{0.6}}{(L)^{0.68}}$$

and $$C = 0.0024(\Delta p_1) - \frac{7.82}{(L)^{0.85}(d)^{1.25}(t - 90)^{1.53}} - \left\{\frac{P_h - 10}{34P_h}\right\}$$

from where the length L, of tube needed of inside diameter d, may be calculated.

Arbitrarily choosing a length of 4 ft of 0.042 in ID

$$\tan \beta = 0.000585(180 + 10)^{0.89} \frac{(0.042)^{0.6}}{(4)^{0.68}}$$

$$= 0.003645$$

$$\delta = 0.0024(0) - \frac{7.82}{(4)^{0.65}(0.042)^{1.25}(170-90)^{1.53}} - \left\{\frac{30 - 10}{34 \times 30}\right\}$$

$$= -0.1751$$

$$\therefore \quad M_s = 140 \times 0.003645 - 0.1751$$

$$= 0.3349 \text{ lb/min}$$

The flow required is 0.523 lb/min, therefore a shorter length is needed,

recalculating for 3 ft length, M_s = 0.4034 lb/min and recalculating for 2 ft length, M_s = 0.5225 lb/min

$\therefore$ the tube is selected 2 ft length of 0.042 in ID.

If the system operates with very wet vapour, more oil will be pumped over. The oil-return capillary, being an unadjustable restriction, will not return, under the given conditions, more than the design value of 0.5225 lb/min of oil-R12 solution. For a short

period oil will build up in the oil separator, reducing the oil level in the compressor. Consequently a decreasing quantity of oil will be pumped over, eventually equal to the quantity of oil returned by the capillary tube. A constant level of oil would be established in the separator.

There is a continuous leakage of refrigerant in solution while the oil is being returned to the compressor, through the capillary tube. This leakage, however, is the same as with the usual type of oil separator incorporating a valve-mechanism.

However, if the system operates below capacity and/or the oil pumped over is less than the design value, a leakage of R12 vapour through the capillary tube will occur. The change in the quantity of oil pumped over is not directly proportional to the change in the cooling load of the system. Assuming that the worst condition, the oil-R12 solution pumped over is half the design value, i.e. 0.262 lb/min.

Considering that other conditions of operation, the discharge pressure and temperature and the suction pressure remain unaltered, i.e. the capacity of the oil-return capillary remains the same, and the oil

pumped over is 0.268 lb/min, the capillary now passes R12 vapour only, for half the period of operation at these conditions.

The R12 vapour leakage is

$$W_G \text{ (lb/min)} = 1.7\ (0.27 \times 0.523)$$

$$= 0.240 \text{ lb/min}$$

where the design capacity of the tube is 0.523 lb/min of oil-R12 solution, the R12 concentration in this solution being 27%.

(Note - It is observed from the experimental results that the mass flow rate of R12 vapour only is approximately 1.7 times the weight of R12 that flows through the tube when flooded with oil-R12 solution.)

This leakage occurs only during 2½ minutes every hour,

∴ the leakage as a percentage of total refrigeration per hour is

$$\frac{0.240 \times 2.5}{6.37(0.95 \times 40 + 0.80 \times 6 + 0.70 \times 5 + 1.1 \times 6)} \times 100$$

$$= 0.165\ \%$$

APPENDIX XII

Tables of Results

TUBE 1 — TABLE N° 1

0.036 IN. INSIDE DIA. 3 FT. LENGTH.

	TEST N°	PRESSURE		M_R	M_0	M_s
		ENTRY	EXIT			
180°F. INITIAL TEMP.	60	160	50	0.077	0.215	0.292
	63	140	50	0.033	0.137	0.170
	64	120	50	0.020	0.090	0.110
	67	100	50	0.015	0.065	0.080
	58	80	50	0.012	0.052	0.064
	61	160	30	0.085	0.227	0.312
	62	140	30	0.039	0.143	0.182
	71	120	30	0.024	0.094	0.118
	72	100	30	0.019	0.068	0.087
	73	80	30	0.014	0.055	0.069
	52	160	10	0.125	0.335	0.460
	53	140	10	0.053	0.179	0.232
	54	120	10	0.030	0.111	0.141
	55	100	10	0.022	0.077	0.099
	56	80	10	0.018	0.059	0.077
150°F. INITIAL TEMP.	74	160	50	0.076	0.160	0.236
	75	140	50	0.043	0.100	0.143
	76	120	50	0.023	0.071	0.094
	77	100	50	0.013	0.059	0.072
	80	160	30	0.080	0.170	0.250
	81	140	30	0.051	0.114	0.165
	82	120	30	0.030	0.081	0.111
	78	100	30	0.017	0.064	0.081
	79	80	30	0.012	0.057	0.069
	83	160	10	0.127	0.175	0.302
	84	140	10	0.076	0.135	0.211
	85	120	10	0.044	0.110	0.154
	86	100	10	0.029	0.097	0.126
	87	80	10	0.022	0.092	0.114
120°F. INITIAL TEMP.	97	120	50	0.059	0.107	0.166
	98	100	50	0.026	0.065	0.091
	99	80	50	0.015	0.040	0.055
	93	120	30	0.065	0.118	0.183
	95	100	30	0.028	0.069	0.097
	96	80	30	0.017	0.044	0.061
	90	120	10	0.071	0.141	0.212
	91	100	10	0.033	0.080	0.113
	92	80	10	0.020	0.053	0.073.

TABLE Nº 2

TUBE 2
0.054 IN. INSIDE DIA. 3 FT. LENGTH

	TEST Nº	PRESSURE ENTRY	PRESSURE EXIT	M_R	M_O	M_S
180°F INITIAL TEMP.	103	160	50	0.162	0.468	0.630
	104	140	50	0.105	0.380	0.485
	105	120	50	0.067	0.312	0.379
	106	100	50	0.040	0.237	0.277
	111	80	50	0.023	0.150	0.173
	106(a)	160	30	0.195	0.512	0.707
	107	140	30	0.125	0.410	0.535
	108	120	30	0.075	0.328	0.403
	109	100	30	0.047	0.251	0.298
	110	80	30	0.029	0.205	0.234
	112	160	10	0.208	0.535	0.743
	113	140	10	0.137	0.423	0.560
	114	120	10	0.083	0.344	0.427
	115	100	10	0.052	0.283	0.335
	116	80	10	0.031	0.230	0.261
150°F INITIAL TEMP.	118	160	50	0.280	0.400	0.680
	119	140	50	0.188	0.320	0.508
	120	120	50	0.105	0.250	0.355
	121	100	50	0.048	0.180	0.228
	122	80	50	0.023	0.125	0.148
	123	160	30	0.297	0.412	0.709
	124	140	30	0.198	0.338	0.536
	125	120	30	0.113	0.270	0.383
	126	100	30	0.058	0.230	0.278
	127	80	30	0.035	0.180	0.215
	128	160	10	0.340	0.425	0.765
	129	140	10	0.215	0.348	0.563
	130	120	10	0.125	0.285	0.410
	131	100	10	0.070	0.232	0.302
	132	80	10	0.045	0.192	0.237
120°F INITIAL TEMP.	133	120	50	0.199	0.216	0.415
	134	100	50	0.108	0.186	0.294
	135	80	50	0.051	0.135	0.186
	136	120	30	0.206	0.218	0.424
	137	100	30	0.112	0.194	0.306
	138	80	30	0.053	0.158	0.211
	139	120	10	0.245	0.233	0.478
	140	100	10	0.135	0.216	0.351
	141	80	10	0.066	0.178	0.244

Tube 3 — Table N°3

0·054 in. Inside Dia. 6 Ft. Length.

	Test N°	Pressure		M_R	M_o	M_s
		Entry	Exit			
180°F Initial Temp.	191	160	50	0·102	0·266	0·368
	192	140	50	0·067	0·235	0·302
	193	120	50	0·039	0·206	0·245
	194	100	50	0·020	0·182	0·202
		80	50			
	195	160	30	0·115	0·300	0·415
	196	140	30	0·079	0·267	0·346
	197	120	30	0·049	0·230	0·279
	198	100	30	0·029	0·196	0·225
	199	80	30	0·016	0·166	0·182
	200	160	10	0·125	0·328	0·453
	201	140	10	0·088	0·286	0·374
	202	120	10	0·057	0·247	0·304
	203	100	10	0·037	0·210	0·247
	204	80	10	0·027	0·180	0·207
150°F Initial Temp.	205	160	50	0·176	0·220	0·396
	206	140	50	0·120	0·210	0·330
	207	120	50	0·072	0·191	0·263
	208	100	50	0·035	0·157	0·190
	209	80	50	0·015	0·115	0·130
	210	160	30	0·185	0·251	0·436
	211	140	30	0·125	0·232	0·357
	212	120	30	0·080	0·208	0·288
	213	100	30	0·048	0·178	0·226
	214	80	30	0·025	0·143	0·168
	215	160	10	0·196	0·260	0·456
	216	140	10	0·131	0·240	0·371
	217	120	10	0·088	0·215	0·303
	218	100	10	0·053	0·185	0·238
	219	80	10	0·032	0·152	0·184
120°F Initial Temp.	220	120	50	0·144	0·172	0·316
	221	100	50	0·069	0·190	0·204
	223	80	50	0·040	0·117	0·157
	224	120	30	0·154	0·190	0·344
	225	100	30	0·080	0·155	0·235
	226	80	30	0·042	0·120	0·162
	227	120	10	0·161	0·197	0·358
	228	100	10	0·091	0·160	0·251
	229	80	10	0·045	0·125	0·170

TUBE 4 — TABLE Nº 4

0.042 IN. INSIDE DIA. 3 FT. LENGTH.

	TEST Nº	PRESSURE		M_R	M_o	M_s
		ENTRY	EXIT			
180°F INITIAL TEMP.	237	160	50	0.149	0.390	0.539
	238	140	50	0.083	0.280	0.363
	241	120	50	0.043	0.205	0.248
	242	100	50	0.021	0.169	0.190
	243	160	30	0.152	0.425	0.577
	244	140	30	0.092	0.302	0.394
	245	120	30	0.052	0.223	0.275
	246	100	30	0.028	0.175	0.203
	247	80	30	0.018	0.142	0.160
	248	160	10	0.162	0.460	0.622
	248(a)	160	10	0.111	0.263	0.374
	249	140	10	0.102	0.320	0.422
	250	120	10	0.060	0.232	0.292
	251	100	10	0.032	0.180	0.212
	252	80	10	0.022	0.150	0.172.
150°F. INITIAL TEMP.	253	160	50	0.182	0.271	0.453
	254	140	50	0.105	0.207	0.312
	255	120	50	0.053	0.159	0.212
	256	100	50	0.023	0.123	0.146
	257	80	50	0.009	0.109	0.118
	258	160	30	0.192	0.281	0.473
	259	140	30	0.117	0.213	0.330
	260	120	30	0.062	0.163	0.225.
	261	100	30	0.033	0.139	0.172
	262	80	30	0.020	0.120	0.140
	263	160	10	0.203	0.291	0.494
	263(a)	160	10	0.173	0.215	0.388
	264	140	10	0.127	0.223	0.350
	265	120	10	0.072	0.176	0.248
	266	100	10	0.043	0.149	0.192
	267	80	10	0.028	0.130	0.158
120°F INITIAL TEMP.	268	120	50	0.142	0.219	0.361
	269	100	50	0.082	0.158	0.240
	270	80	50	0.028	0.099	0.127
	271	120	30	0.151	0.222	0.373
	272	100	30	0.088	0.164	0.252
	273	80	30	0.033	0.107	0.140
	274	120	10	0.155	0.225	0.380
	275	100	10	0.093	0.170	0.263
	276.	80	10	0.050	0.116	0.166.

Tube 5

Table N° 5

0.042 in. Inside Dia. 6 ft. Length.

	Test N°	Pressure		M_R	M_o	M_s
		Entry	Exit			
180°F. Initial Temp.	142	160	50	0.081	0.230	0.311
	143	140	50	0.053	0.193	0.246
	144	120	50	0.034	0.155	0.189
	145	100	50	0.022	0.131	0.153
	146	160	30	0.086	0.245	0.331
	147	140	30	0.059	0.205	0.264
	148	120	30	0.039	0.167	0.206
	149	100	30	0.025	0.121	0.146
	150	80	30	0.014	0.095	0.109
	151	160	10	0.089	0.251	0.340
	152	140	10	0.058	0.198	0.256
	152(a)	140	10	0.061	0.212	0.273
	153	120	10	0.043	0.174	0.217
	154	100	10	0.028	0.140	0.168
	155	80	10	0.018	0.108	0.126.
150°F. Initial Temp.	156	160	50	0.131	0.200	0.331
	157	140	50	0.076	0.172	0.248
	158	120	50	0.046	0.142	0.188
	159	100	50	0.028	0.112	0.140
	161	160	30	0.135	0.204	0.339
	162	140	30	0.080	0.175	0.255
	163	120	30	0.050	0.145	0.195
	164	100	30	0.033	0.115	0.148
	165	80	30	0.027	0.083	0.110
	166	160	10	0.146	0.210	0.356
	167	160	10	0.128	0.183	0.311
	168	140	10	0.087	0.180	0.267
	169	120	10	0.055	0.150	0.205
	170	100	10	0.041	0.119	0.160
	171	80	10	0.037	0.089	0.126.
120°F. Initial Temp.	172	120	50	0.119	0.122	0.241
	173	100	50	0.052	0.095	0.147
	174	120	30	0.133	0.138	0.272
	175	100	30	0.057	0.110	0.167
	176	80	30	0.029	0.095	0.124
	177	120	10	0.147	0.155	0.302
	181	100	10	0.065	0.115	0.180
	182	80	10	0.033	0.100	0.133

TABLE No 6

COMPARISON OF ACTUAL CONCENTRATION WITH THOSE GIVEN BY THE SOLUBILITY CURVES FOR OIL-R12 SOLUTIONS, FIG. 50

INITIAL		PERCENTAGE R-12 CONCENTRATION		
TEMP °F	PRESS (PSIG)	MEASURED VALUE (AVERAGE)	SOLUBILITY CURVES FIG. 50.	DIFFERENCE
180	160	27	28	-1
180	140	22·2	23·7	-1·5
180	120	17·7	19·4	-1·7
180	100	15·0	16	-1
180	80	13·2	12·5	+0·7
150	160	41·2	43	-1·8
150	140	37	36	+1
150	120	29·6	29·2	+0·4
150	100	21·2	23	-1·8
150	80	16	17·8	-1·8
120	120	48	50·8	-2·8
120	100	36·7	37·6	-0·9
120	80	27·4	27	+0·4

TABLE No 7

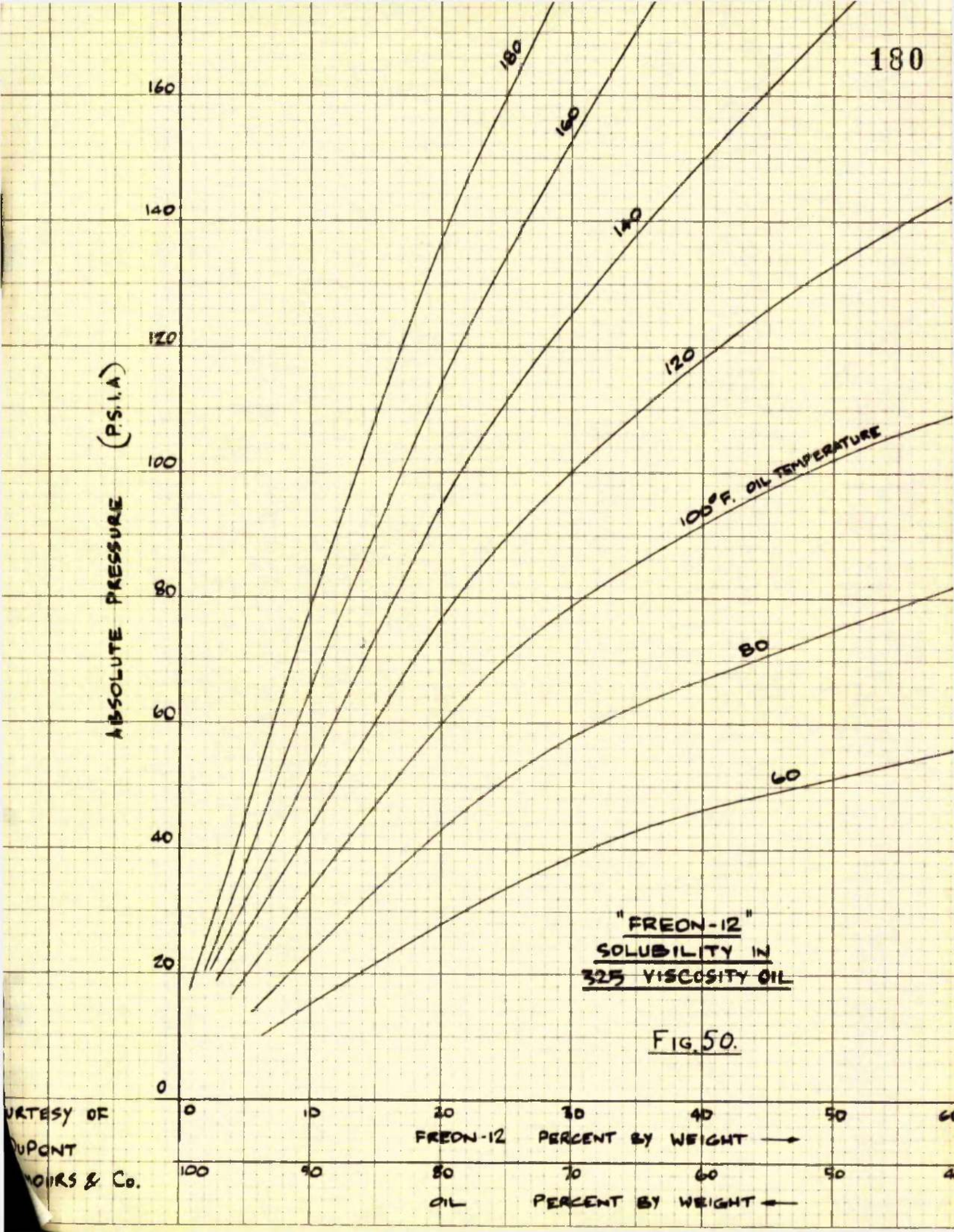
COMPARISON OF THE COMPONENTS OF OIL-R12 SOLN WITH INDIVIDUAL FLOWS OF OIL AND R12

TUBE 4, 0·042 IN ID, 3 FT LENGTH, TEMP. - 150°F

PRESSURE		COMPONENT OF SOLUTION	EXPERIMENTAL	CALCULATED
HEAD	BACK	OIL		
(PSIG)	(PSIG)	M_o (lb/min)	M_{oe} (lb/min)	M_{oc} (lb/min)
160	10	0·291	0·091	0·124
140	10	0·223	0·076	0·107
120	10	0·176	0·063	0·091
100	10	0·149	0·058	0·074
80	10	0·130	0·049	0·058
HEAD	BACK	REFRIGERANT 12		
(PSIG)	(PSIG)	M_r (lb/min)	M_{ge} (lb/min)	M_{gc} (lb/min)
160	10	0·203	0·216	0·225
140	10	0·127	0·182	0·164
120	10	0·072	0·157	0·119
100	10	0·043	0·133	0·082
80	10	0·028	0·166	0·054

APPENDIX IV

Data Graphs Used



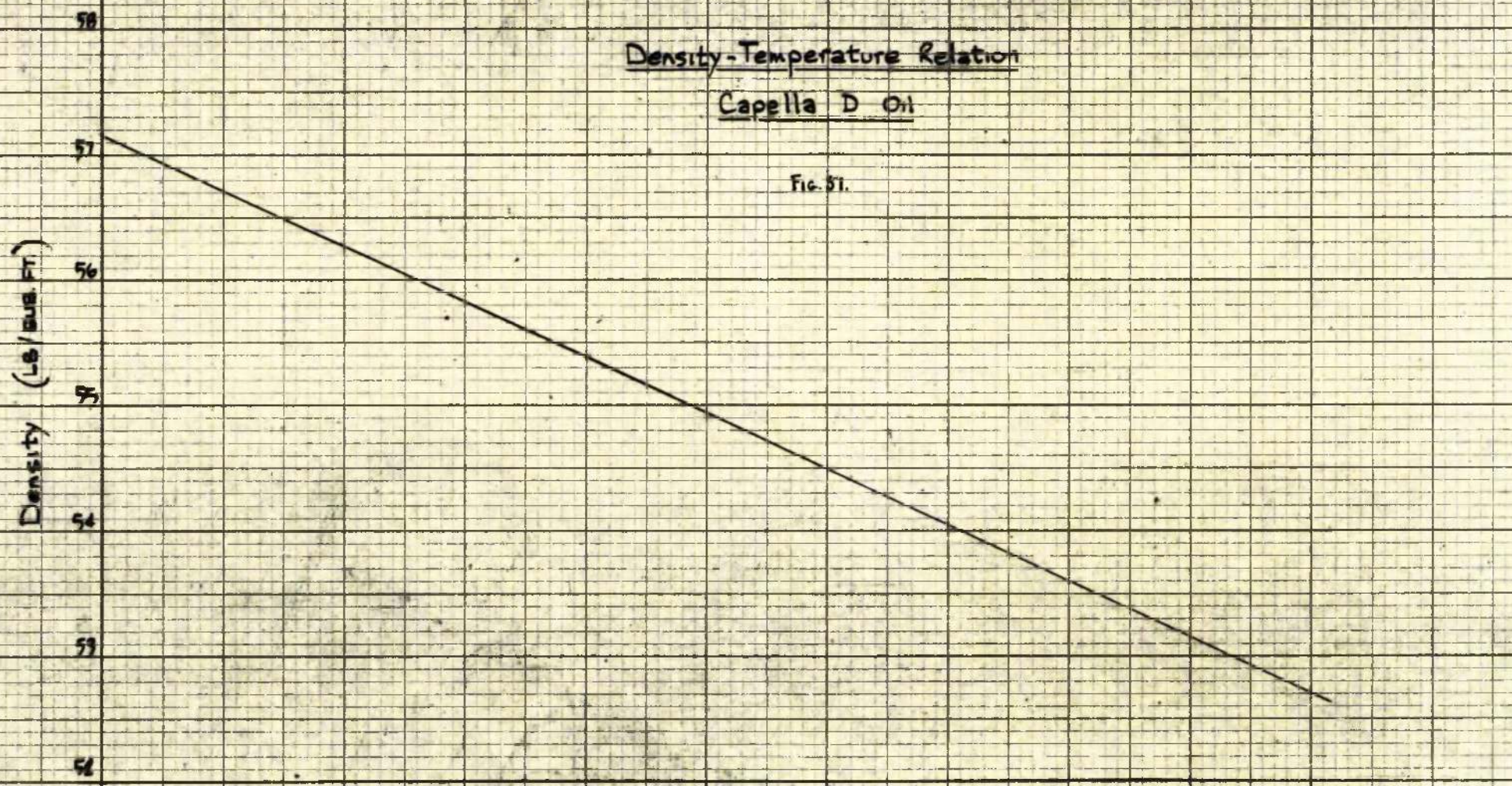
"FREON-12"
SOLUBILITY IN
325 VISCOSITY OIL

FIG. 50.

Density - Temperature Relation

Capella D Oil

Fig. 51.



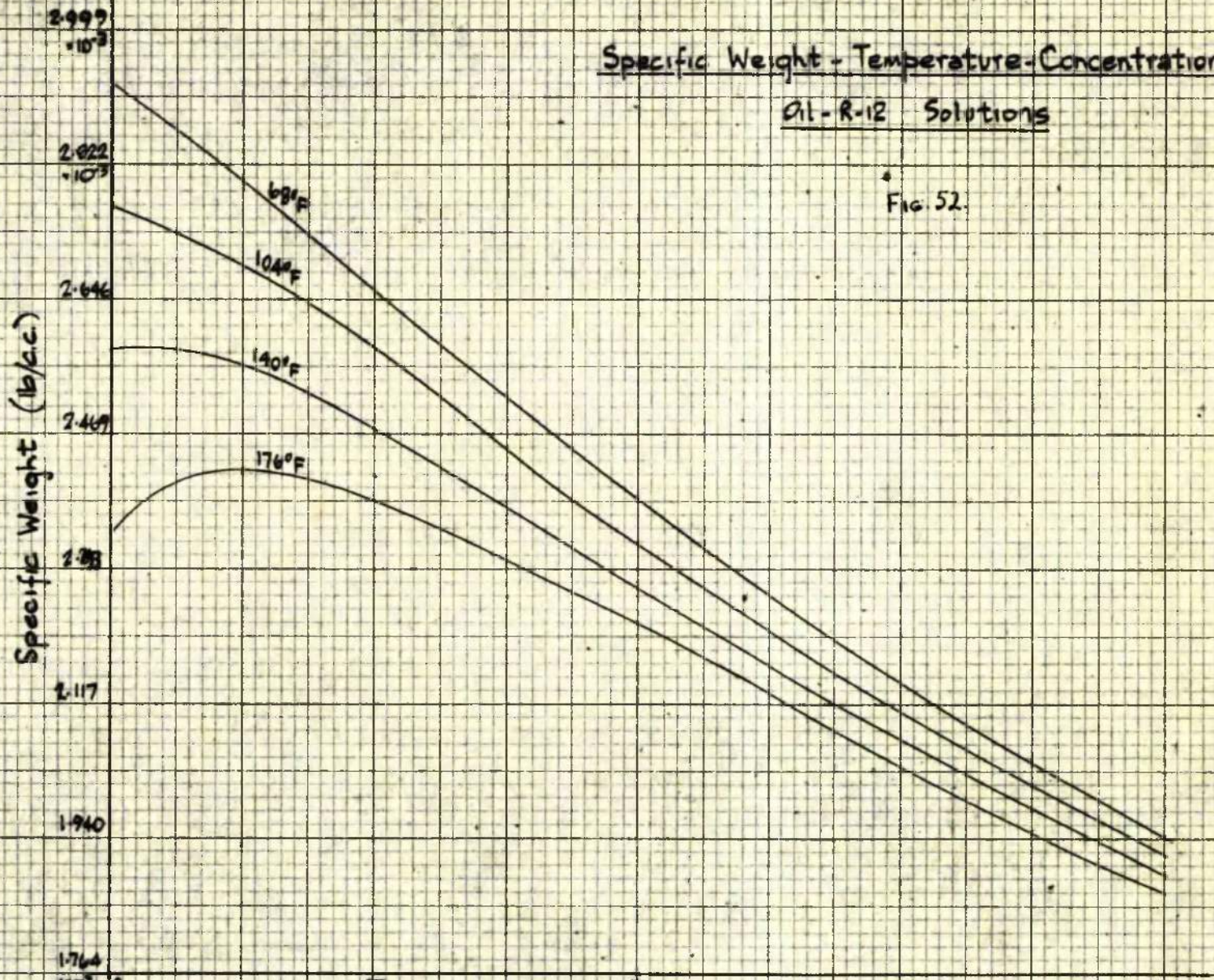
By Courtesy of Regent Oil Co. Ltd

FIG. 51.

Specific Weight - Temperature - Concentration Relation

Oil - R-12 Solutions

Fig. 52.



Reproduced from "Density of Oil-Refrigerant Mixtures" by H.J. Löffler (21)

Fig. 52.